ENGINEERING WORKS PRACTICE

General Editor. E. MOLLOY

Advisory Editor: F. A. NORTON, C.B.E., M.Sc., M.I.Mech.E., F.R.Ae.S., M.Inst.T.

VOLUME IV

Installation, Operation and Maintenance

Contributors:

J. R. FAWCETT, B.Sc. (Hons.), A.M.I.Mech.E.

A. E. WILLIAMS, Ph.D., F.C.S.

E. MOLLOY

J. CHAMBERLAIN

W. LAMB, M.I.Mar.E., M.Inst.F.

T. D. WALSHAW, B.Sc. (Eng.), D.L.C. (Hons.), A.M.I.P.E., A.M.I.Mech.E.

R. SHILTON, B.Sc. (Eng.) (Hons.), A.M.I.Mech.E., A.M.I.P.E., A.M.I.A.A.

E. D. EVANS, M.S.W.I.E.

D. L. PERRY

E. PULL, M.I.Mech.E., M.I.Mar.E.

G. W. WILLIAMSON, O.B.E., M.C., M.Inst.C.E., M.I.Mech.E., F.R.Ae.S.

D. J. SMITH, O.B.E., M.I.Mech.E.

J. H. S. YOUNG

LONDON

GEORGE NEWNES LIMITED

TOWER HOUSE, SOUTHAMPTON STREET

STRAND, W.C.2

1912

A Consolidated Index and Classified Key, enabling references to any subject to be readily found, are given at the end of this Volume

PRINTED AND BOUND IN ENGLAND BY
HAZELL WATSON AND VINEY LTD
AYLESBURY AND LONDON

CONTENTS OF VOLUME IV

CONTENTS

INSTALLATION, OPERATION AND MAINTENANCE

INSTRUMENTS IN MECHANICAL ENGINEERING

INSTRUMENTS are the senses of industry and are used to augment the human senses and to detect, measure, record, and control the many phenomena encountered in manufacturing processes and in the operation of machinery. Of these, the most important are pressure and vacuum, temperature, speed, volume, weight, and flow. Instruments are also used for inspecting materials and conducting tests of strength and endurance. The tendency to-day is to mount instruments on panels in a position for easy observation, and many types are designed for this purpose. The power-station instrument panel (Fig. 1) is a good example of the centralisation of instruments. Included are flowmeters, pressure gauges, electrical thermometers with selector switches, pressure thermometers (two fitted with electric contacts), and in the centre a U-column gauge to show the absolute pressure in the condenser.

Many machines and processes are now controlled entirely by instruments which open and close valves and switches, thus improving both quality and efficiency of operation and avoiding much tedious manipulation.

PRESSURE

Pressure is usually measured in pounds per square inch (p.s.i. or lb./in.^{2}) above that of the atmosphere, which exerts a pressure of 14·7 p.s.i. approximately. A pressure below atmospheric is called a vacuum, and is measured in inches of mercury. Low pressures and draughts are given in inches of water. A simple pressure gauge is the U-tube manometer, which measures pressure by the difference of level between the liquid in the two limbs. For general use the single-column gauge (Fig. 2) is preferred. The pressure is applied to the cistern and read off directly on the scale. For vacuum readings the connection is made to the top of the column. The cup at the top prevents the liquid spilling if too much pressure is applied. This design is simple and reliable, and is used for pressures up to 20 p.s.i. when filled with mercury and to 1½ p.s.i. when water or oil filled.

The Bourdon Gauge

The dial gauge (Fig. 3) is used for measuring steam and air pressures, the Bourdon tube being made of phosphor bronze for pressures up to 1,000 p.s.i. For higher pressures, up to a maximum of 40 tons/in.^{2}, the Bourdon tubes are made of heat-treated beryllium copper or spring steel. Gauges for use with ammonia and other substances which attack copper alloys may have all contact parts made of steel.

The tube, which is of oval section, tends to straighten when pressure is applied, so that the free end lifts in proportion to the pressure. It is connected to the pointer by a fine-toothed magnifying gear. Like most modern instruments, pressure gauges are made in types suitable for flush panel mounting, surface mounting (as shown), or direct mounting.

Diaphragm Gauges

Bourdon gauges are not recommended for pressures below 15 p.s.i., and are rarely made for pressures below 5 p.s.i., so that diaphragm or capsule gauges are often used. These are less liable to damage through misuse, and diaphragm gauges can be made in special patterns for handling corrosive chemicals, withstanding pressure, and dealing with liquids which would clog an ordinary gauge.

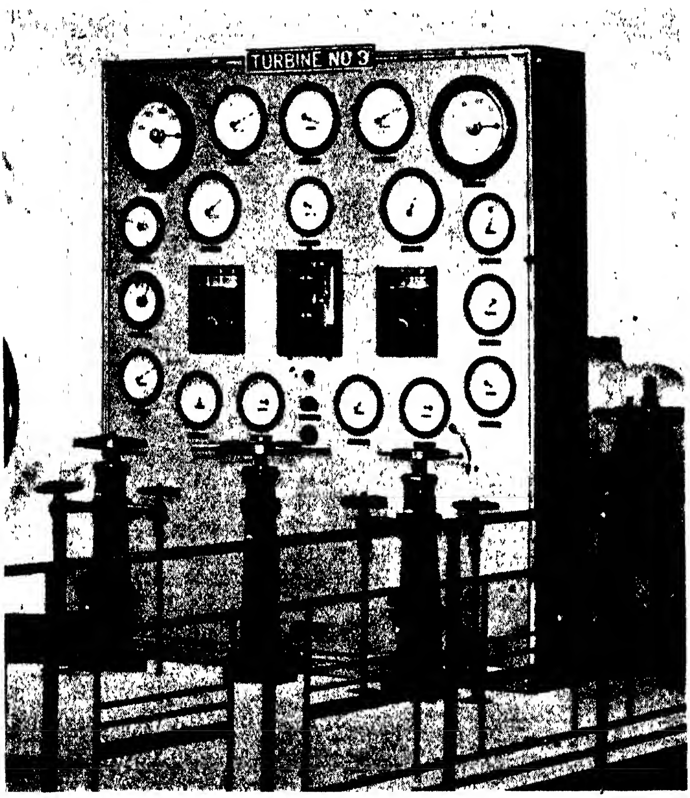


FIG. 1.—POWER-STATION INSTRUMENT PANEL (*George Kent, Ltd.*)

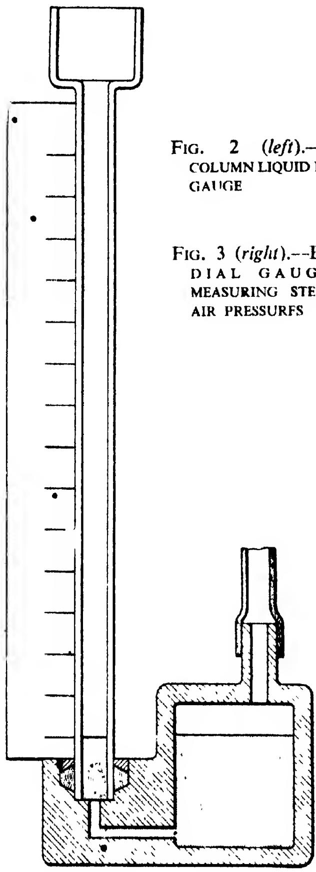


FIG. 2 (*left*).—SINGLE-COLUMN LIQUID PRESSURE GAUGE

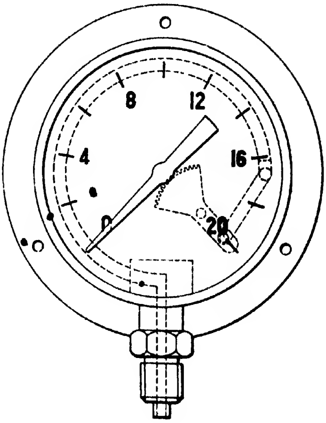


FIG. 3 (*right*).—BOURDON DIAL GAUGE FOR MEASURING STEAM AND AIR PRESSURES

The altimeter (Fig. 4) is an example of a capsule gauge. The twin aneroid capsules are completely evacuated and sealed off. As the altitude increases, the atmospheric pressure inside the case decreases and the capsules distend and rotate the pointers through the magnifying gearing. Three pointers show hundreds, thousands, and tens of thousands of feet respectively, and are geared together like the hands of a clock. Because of the great magnification the movement is jewelled throughout to reduce friction. By setting the counter to the ground atmospheric pressure in millibars, the altimeter shows the height over the aerodrome.

Installation

Pressure gauges must not be screwed directly into steam lines, but should be protected by a water seal formed by a proprietary siphon or a length of pipe to prevent live steam reaching the gauge. Gauges mounted on panels may be above or below the tapping point, but if so, allowance must be made for the head of liquid in the connecting pipe during calibration. When fitted to hydraulic lines, protection against pressure shocks should be given by fitting a needle valve or other protective device.

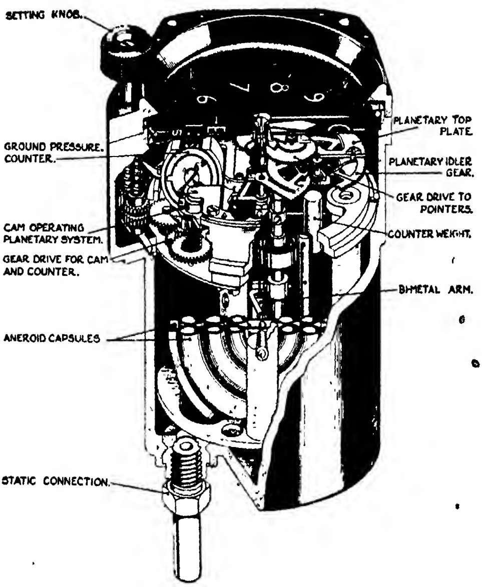


FIG. 4.—ALTIMETER (*Smiths Aircraft Instruments, Ltd.*)

TEMPERATURE

Correct temperature measurement is essential for the efficient running of all types of power plants, for the heat treatment of metals and in many other processes. In this country, the Fahrenheit scale is in general use (water freezes at 32° F. and boils at 212° F.), but for scientific work the Centigrade scale (water freezes at 0° C. and boils at 100° C.) is used. Temperatures for the heat treatment of metals are also given in Centigrade. The temperatures used in industry range from − 200° C. for deep freezing to 3,400° C. for molten tungsten. Glass thermometers are simple and reliable, and are used for temperatures down to − 100° C. and up to + 450° C., but

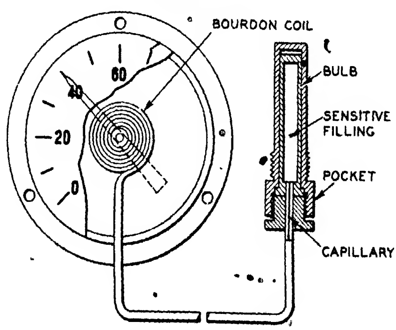


FIG. 5.—PRESSURE-ACTUATED THERMOMETER

are fragile and difficult to read from a distance. Dial thermometers are much more easily read, can be mounted up to 150 ft. away from the bulb, and cover a range of − 20° C. to + 600° C. Mercury-in-steel and vapour thermometers (Fig. 5) are pressure operated, the bulb being connected to a special form of pressure gauge by a capillary tube enclosed either in a flexible sheathing, which can be laid like electric cable, or for direct mounting in a short rigid stem.

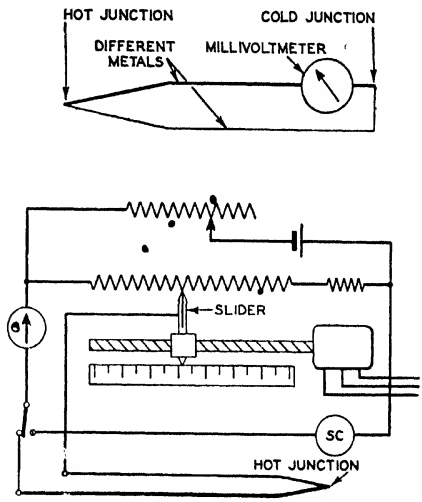


FIG. 6.—THERMOCOUPLE PYROMETERS

Pressure-actuated Thermometers

Mercury-in-steel thermometers have steel bulbs, capillary tubes and coils, and are filled with mercury under high pressure. The expansion of the mercury in the bulb causes an increase in pressure, and this operates the coil and pointer. The effects of temperature variations on the capillary and coil are minimised by using a capillary with a very fine (0·005 in.) bore and by bi-metallic compensators. Vapour thermometers are made of non-ferrous metal, and are partially filled with a volatile substance, such as ether or methyl chloride. The increase of vapour pressure with temperature operates the pointer. The scale is non-uniform, and is easiest to read at the higher temperatures, and this type is most useful when temperatures are fairly constant. Mercury thermometers have uniform scales, and are best for general use up to 500° C. Bulbs in pressure pipes and steam lines are protected by a pocket of suitable material.

Bi-metallic Dial Thermometers

Bi-metallic dial thermometers have a helix of bi-metal in the bulb which twists with changes of temperature and rotates the pointer either direct or through gearing. The bulbs of dial thermometers are normally fitted with a $\frac{3}{4}$-in. B.S.P. union connection, and the sensitive portion is $\frac{1}{2}$–$\frac{5}{8}$ in. diameter and 3 in. long. For accuracy the bulb must be located at a representative point and the whole of it thoroughly heated. Pockets are used for bulbs inserted in pipes under pressure to protect them and enable them to be taken out without releasing pressure.

Thermocouple Pyrometers

A thermocouple (Fig. 6) is formed when two wires of different metals are joined together and one junction is heated. The E.M.F. (voltage) produced is measured by a millivoltmeter or potentiometer. The instrument is calibrated in degrees, and may have automatic compensation for variations in cold junction temperature. For furnaces, couples are made of 100 per cent. platinum/90 per cent. platinum, 10 per cent. rhodium. The potentiometer measures the E.M.F. by comparing it with a standard cell. Manual potentiometers are used for testing. The automatic type shown has motor-driven balancing mechanism, and records a number of temperatures by an automatic selector switch.

Although thermocouples will measure temperatures from − 200° C. to + 3,000° C., in engineering they are mainly used over the range 500° to 1,450° C. for hardening and tempering furnaces and to 3,000° C. for molten metals. For laboratory work, unprotected thermocouples are used because of their great sensitivity, and surface temperatures can be taken very accurately with thermocouples made of thin strip pressed on to the surface. Couples for industrial use are usually protected by substantial refractory sheaths to avoid corrosion. Indicators fitted to heat-treatment furnaces are of the edgewise type, but large circular dials (Fig. 7) are preferred for molten-metal readings. Most recorders

FIG. 7.—MEASURING THE TEMPERATURE OF MOLTEN STEEL
(*Tinsley Industrial Instruments, Ltd.*)

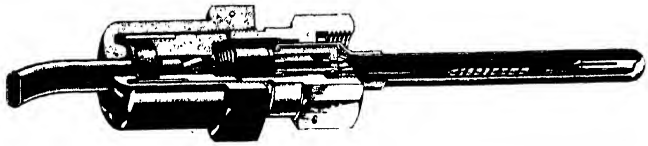


FIG. 8.—RESISTANCE THERMOMETER BULB

are of the potentiometer type, and record a number of temperatures simultaneously.

Resistance Thermometers

In this type of thermometer (Fig. 8), the brass sheath soldered into a screwed socket contains a resistance element which is sensitive to temperature changes, and a non-temperature-sensitive resistance for calibrating to a standard. The bulb is connected by leads to a ratiometer indicator which compensates for lead errors. Current is supplied from the mains through a transformer, rectifier, and barretter, or from a battery. Resistance thermometers are used on aircraft for air, water, and oil temperatures, and are preferred to the pressure type, because it is easier to install and maintain a cable which can be disconnected at both ends than a capillary which cannot be detached from the instrument. If desired, a number of bulbs can be connected to one indicator through a selector switch, so effecting an appreciable economy in first cost. The platinum resistance thermometer used in conjunction with an accurate resistance box is an extremely accurate instrument, and forms the basis of the international temperature scale.

Optical and Total Radiation Pyrometers

Bodies and furnaces at temperatures above 500° C. emit light, and it is possible to measure the temperature optically without making direct contact. The readings may not be so accurate as by contact methods, but the inaccuracies in any particular application are often constant and can be allowed for. Optical pyrometers have a telescope which is sighted on the object or furnace wall, and the operator judges when the colour is the same as that of the filament of a lamp in the instrument and controlled by a resistance. A dial gives the corresponding temperature.

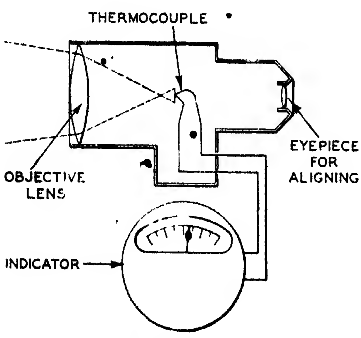


FIG. 9.—TOTAL RADIATION PYROMETER

In the total radiation pyrometer (Fig. 9) the radiant heat from the furnace is focused on to a thermo-

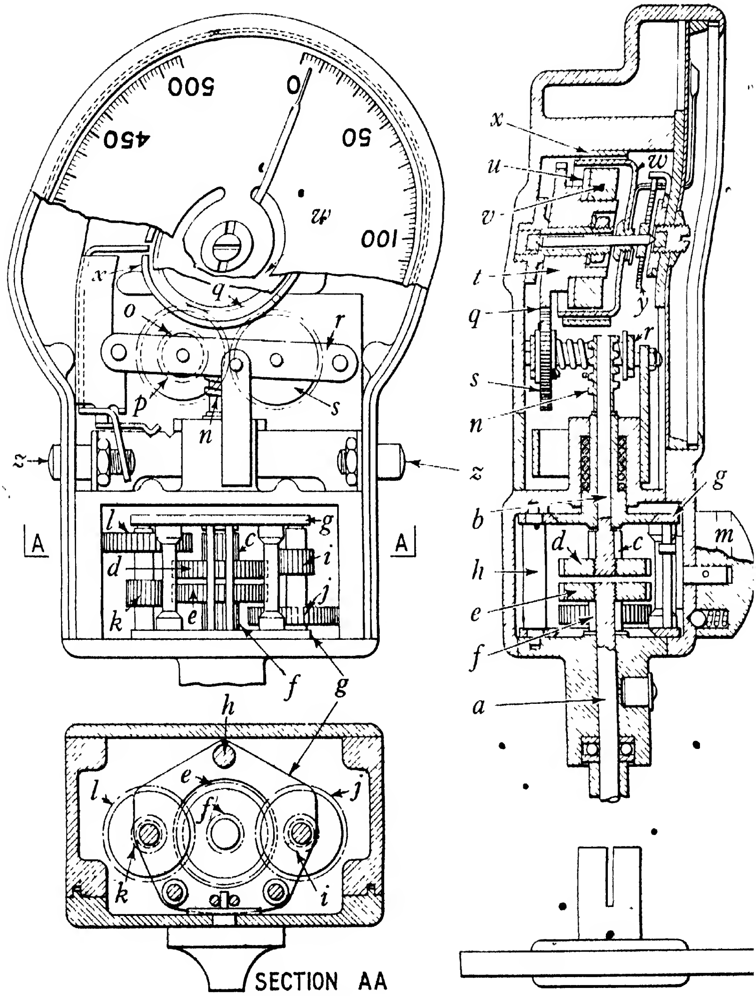


FIG. 10.—HAND TACHOMETER (*Smiths Industrial Instruments, Ltd.*)

couple or photocell by the objective lens, and the current produced is shown on a meter as the temperature. These pyrometers give continuous readings and can be used for control purposes. For permanent installation the case can be water-cooled. Both these pyrometers are suitable for molten metals, glass furnaces, rolled sections, refractories, etc. The distance is not important so long as the image completely fills the lens.

SPEED

Speed Indicators (Tachometers)

The magnetic drag tachometer (Fig. 10) is widely used as a car speedometer, for aircraft engine speed indicators, and for hand tachometers. The drive from the main spindle (*a*) is taken through a three-speed gearbox operated by an external knob (*m*) to a worm drive. The drive is then taken through a gear train (*o*, *p*, and *r*), which gives a unidirectional drive whatever the direction of rotation of the main spindle (*a*). The instrument pointer is attached to an aluminium cup (*w*), which is affected by the rotary magnet (*v*), the eddy currents produced tending to turn the cup against the resistance of the hairspring (*y*). The strength of the hairspring is chosen so that the pointer indicates the correct speed. The knobs (*z*) when pressed hold the pointer from returning to zero and allow the reading to be taken after the tachometer has been withdrawn.

Car speedometers are driven by a flexible shaft, but on aircraft and locomotives, where the distance is too great for this, a three-phase generator driven by the engine or wheels is connected to a synchronous motor mounted in the back of the tachometer case.

The pendulum-type tachometer has a weighted pendulum fitted to the rotating spindle which flies out under centrifugal force and so operates the pointer. Pendulum tachometers, unlike magnetic ones which commence to read at zero, are only suitable for a restricted speed range, but because of their more open scale, are preferred for constant-speed turbines and engines. Tachometers and generators may be driven direct from the end of a cam or similar shaft, or by flexible shaft or belt, whichever method is most convenient.

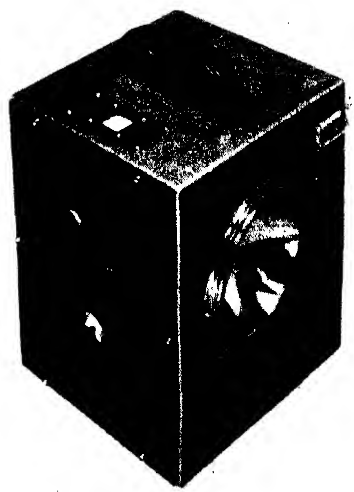


FIG. 11.—STROBOSCOPE (*Dawe Instruments, Ltd.*)

The Stroboscope

An electronic controller causes a discharge lamp to give a succession of flashes of a few micro-seconds

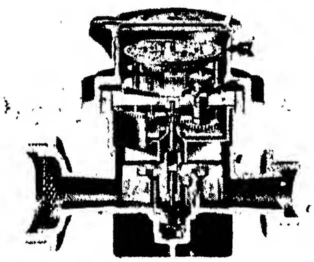


FIG. 12.—INFERENTIAL WATER METER
(*Leeds Water Meter Co., Ltd.*)

duration. The number of these flashes can be read off on a dial. If the stroboscope lamp is shone on to moving machinery, when the number of flashes per minute are the same as the speed of the machine, the moving parts will appear to be stationary. When the speed of flashing is varied slightly, the machine parts appear to be moving slowly, and the effects of stresses, bounce, and vibration can readily be observed.

In the stroboscope illustrated in Fig. 11, the case contains a multi-vibrating thermionic valve circuit connected to the discharge lamp let into the front. The frequency is controlled by a potentiometer operated by the upper knob, the setting being shown on the scale above. To check a speed the frequency is gradually *reduced* until the moving parts appear to be stationary; they will also appear stationary if the frequency is a submultiple ($\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, etc.) of the speed. By slightly varying the frequency, a mechanism can be examined in slow motion.

VOLUME AND FLOW

It is now accepted that for efficiency and economy all fluids used in bulk should be metered both where they are produced and where they are used. Small meters, such as those used for gas, water, and liquid fuels, have a piston or bellows which is moved by the fluid passing through the meter. The action is positive and the readings accurate, but the capacity is limited, nor is it possible to use this type for hot fluids. The quantity metered may be shown on a dial attached to the meter; in aircraft the amount of fuel used is recorded on the pilot's instrument panel by a counter which is actuated by a switch on the meter at each tenth gallon.

Inferential Meters

Inferential meters have a turbine or helical vane in the fluid stream which is rotated like a propeller and geared to a counter. They are not so accurate as the piston type, but can be used for steam and flows of any magnitude. An inferential meter is illustrated in Fig. 12. The turbine is rotated by the water passing through the meter, the number of revolutions varying with the volume of flow. The lower gear operates in water, and the movement is then transmitted through a gland to the dial gear, which is in air. A meter of this type will handle water at a pressure of 200 lb. per square inch and temperature of 300° F.

Differential Meters

The differential pressure flowmeter (Fig. 13) which measures the pressure drop as a Venturi tube or orifice plate in the flowstream is of universal application. The meter can be mounted away from the pipe, there are no moving parts in the fluid, nor is there any limitation as to the nature of the fluid. The Venturi tube forms a constriction in the pipe, which increases the velocity and so lowers the pressure. Tappings are led to a differential pressure gauge, which consists of a steel U-tube containing mercury. A float on top of the mercury is connected through a pressure-tight gland to a recording pen or counting mechanism. The flow varies with the square root of the differential pressure, and either the chart is calibrated to suit or a cam mechanism converts the reading to a straight-line law. A Venturi tube has a short approach taper and a long downstream taper and causes negligible loss of head. The orifice plate gives similar results, and is considerably cheaper to make and install, but it causes a loss of head equal to approximately 50 per cent. of the differential pressure. The meter itself (Fig. 14) is usually of the recording type, and the flow is recorded continuously on the clockwork-driven chart, and can also be totalised by fitting an integrating

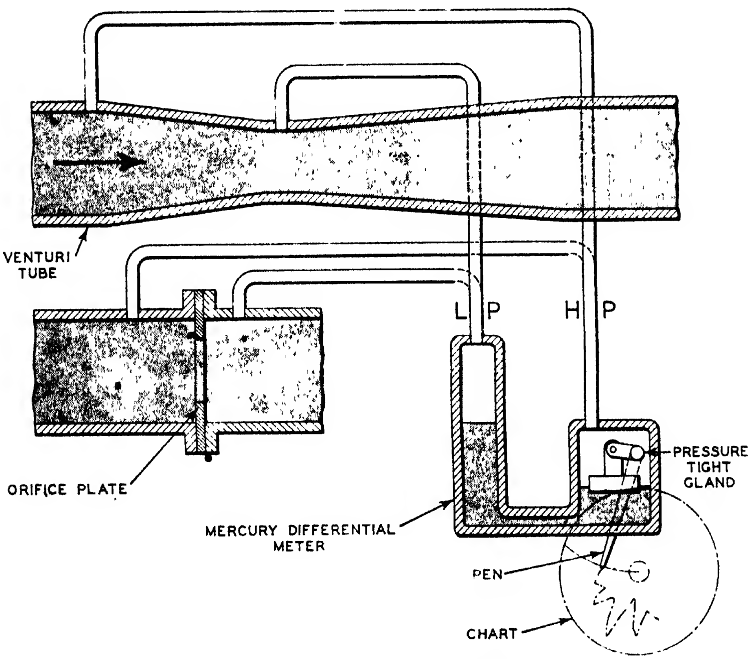


FIG. 13.—DIFFERENTIAL FLOWMETER

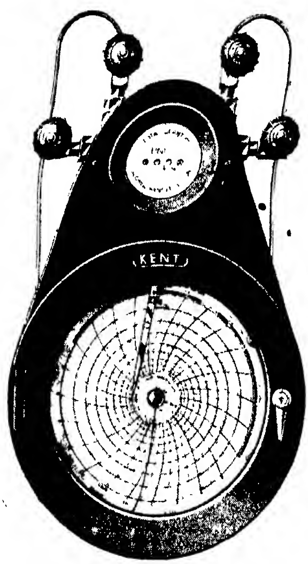


FIG. 14.—FLOWMETER
(*George Kent, Ltd.*)

mechanism. Meters for gas and steam give a pressure record also, as the amount passed varies with the pressure as well as with the velocity.

Installation of Flowmeters

Flowmeters must not be subjected to rapid fluctuations of flow, as these can cause serious inaccuracies. Meters should be installed as far away from valves, bends, and side branches as possible, and there should be at least fifteen diameters of straight pipe either side. Differential meters should be below the pipe if possible, and provision should always be made for bleeding the connection tubes between pipe and meter, as the presence of even a small quantity of air will cause serious inaccuracies.

Pitot Tube

This has a tube facing upstream, and either a similar tube facing downstream or a shielded tube to give the static pressure. The rate of flow varies as the square root of the difference of the pressure in the two pipes. Pitot tubes are occasionally used for measuring the velocity of fluids in pipes, but nowadays their chief use is as air-speed indicators in aircraft. As it is impracticable in aircraft to use a double tube, the pitot tube and static tube are incorporated in one unit (Fig. 15) and connected to a sensitive differential gauge calibrated in miles per hour or knots. The static tube is connected to the inside of the case and the pressure tube to the diaphragm. The difference of pressure is shown as the speed.

Liquid Level Measurement

The level of the liquid in tanks (Fig. 16), reservoirs, etc., can be measured directly or, as is often more convenient when the tank is inaccesssible, shown at a distance on a dial gauge. A simple method is to fix a gauge glass, similar to a boiler water-level gauge, to the side of the tank (*A*), but clearer readings are given with some form of dial gauge such as the

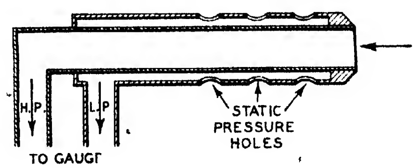


FIG. 15.—PITOT TUBE

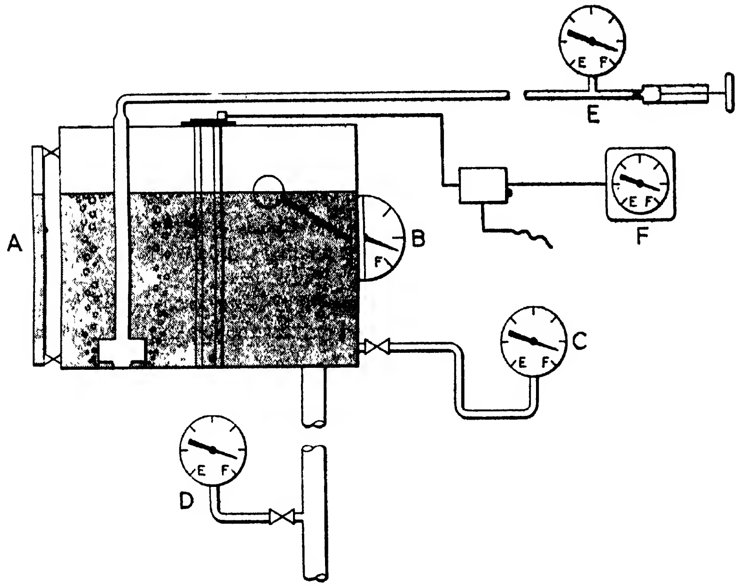


FIG. 16.—MEASURING LEVEL OF LIQUID IN A TANK

R. & G. Fluid Measure (Fig. 17). The float is pivoted on a spindle, which operates the pointer through a rotating magnet, no stuffing box being necessary.

A Bourdon pressure gauge fitted level with or slightly below the tank bottom will indicate the height of liquid, and can be mounted at a distance (Fig. 16 *C*). Special Bourdon gauges which allow for the head of water in the downpipe are suitable for overhead tanks such as power-station surge tanks (Fig. 16 *D*). Pneumatically operated pressure systems can be used with the dial up to 150 ft. away from the tank: a vertical pipe is fixed in the tank, with its open end near the bottom and its upper end connected to a pressure gauge. Air is then pumped into the system until it bubbles out at the end of the pipe. The gauge, suitably calibrated, will show the level of liquid (Fig. 16 *E*). A similar system dispenses with the pump, and uses a very flexible rubber diaphragm to seal the ends of the dip pipe. Float gauges give the volume of liquid, whilst pressure-operated tank gauges give equal increments for equal weights of liquid, the specific gravity of which must be known if the volume is required. A description of an electronic tank gauge (Fig. 16 *F*) is given on page 27.

AUTOMATIC CONTROL

By controlling automatically such factors as pressure, temperature, flow, speed, liquid level, specific gravity, *p*H, humidity, and time, greater efficiency

and economy are often obtained, whilst tedious manual operation with its lack of accuracy is eliminated. There are several methods of automatic control in general use. Four are illustrated in Fig. 18:

A. ON/OFF.—The temperature fluctuates between two extremes, the frequency of operation of the switch depending on the load.

B. PROPORTIONAL.—A change of load causes a corresponding shift from the set value, and this in turn causes the valve or damper opening to be adjusted, but the set value is only attained for the standard load.

C. PROPORTIONAL PLUS FLOATING.—The addition of the floating control makes it possible to restore the value to the original setting whatever the loading, short of an overload.

D. PROPORTIONAL PLUS FLOATING PLUS FIRST DERIVATIVE.—This takes into account the rate of change of a value, and reduces the tendency to overcorrect, so, in practice, giving closer control than *C*.

Most simple controllers, especially electrical ones, are of the on/off type, and are suitable where no great accuracy is required and where there is a reservoir of energy which can be drawn on during the "off" period. They are used for controlling electric furnaces, air compressors, pumps for filling tanks, etc. When energy must be supplied continuously, as for example to a steam turbine, a more flexible control is needed, and pneumatic or hydraulic controllers are used. For proportional control the steam valve is opened in proportion to the decrease of speed caused by an increase of load, so that the speed is continually changing as the load varies, and if there is a permanent increase of load, the speed will be reduced unless the governor is reset. This is done automatically in the proportional and floating system which superimposes a tendency to correct any departure from the control point. In practice there are many factors which cause control to be erratic, and it is to deal with these that the more complicated systems, such as proportional, floating, and derivative, are used.

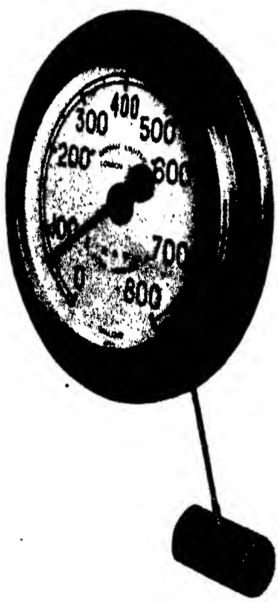


FIG. 17.—FLOAT-OPERATED TANK DIAL GAUGE (*Bayham, Ltd.*)

Thermostats

Simple thermostats have a bi-metallic element in rod or strip form which operates a switch or gas valve as the temperature changes. Rod-type elements are of limited application, and are mainly used for water heaters. Strip elements are very compact and sensitive, and can be incorporated in the mechanism to give protection against over-

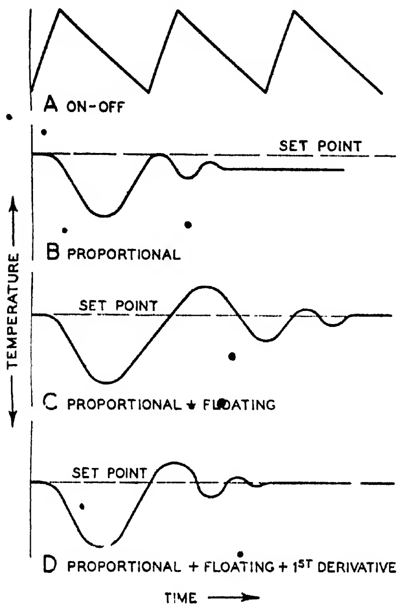


FIG. 18.—AUTOMATIC CONTROL METHODS

heating. For general use, a thermostat operated by a mercury-in-steel or vapour thermometer element is preferred, as this enables a bulb to be located in the best position with the thermostat itself mounted visibly.

The indicating thermostat shown in Fig. 19 indicates the temperature and also operates a 12-amp. A.C. 250-v. switch at the temperature shown by the setting pointer. This is suitable for controlling plastic moulding presses, refrigerator motors, etc. With other instruments incorporating mercury switches, currents up to 20 amps. A.C. or D.C. can be handled. A thermocouple-actuated indicator, incorporating a switch which cuts off when the set point is reached, is used for heat-treatment furnace control. As the millivoltmeter movement is not powerful enough to work a switch direct, it is necessary to fit either a motor-driven trip or an electronic amplifier actuated by a vane on the pointer. Gas and oil supplies are frequently controlled by a magnetic valve working with an electrical thermostat.

Non-electrical thermostats for controlling steam have a bulb connected to a metallic bellows which is partially filled with a volatile liquid. The bellows is connected to the valve stem and the variation of vapour pressure with temperature opens or closes the steam valve. An instrument of this type is illustrated in Fig. 20, which shows the bulb on the right of the valve. Its operating temperature is adjustable over a fixed range, various ranges being obtainable. The difference in temperature between fully open and fully closed is approximately 9° F., but this varies with the size of valve and operating temperature range.

FIG. 19.—INDICATING THERMOSTAT
(*The British Thermostat Co., Ltd.*)

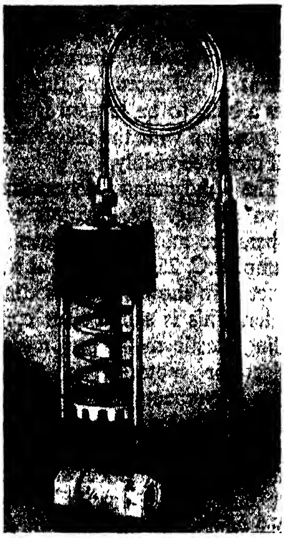


FIG. 20.—THERMOSTATIC VALVE
(*The British Thermostat Co., Ltd.*)

Pressurestats

Reducing valves give a low-pressure supply of gas or steam from a high-pressure source. They are operated by a diaphragm connected to the low-pressure side and loaded by a spring or weight to give the required pressure. Demand causes the pressure to drop slightly, so that the diaphragm moves under the load and opens the valve until the pressure is restored.

The control of air compressors and pumps and the operation of low- and high-pressure alarms is often carried out with pressure gauges fitted with electric contacts (Fig. 21) operating through relays if the current exceeds 0·5 amp. Pressures from 1 in. of water to 6 tons per square inch can be dealt with to an accuracy of 1 per cent. Bellows-operated pressurestats are used for certain specialised purposes, such as stopping and starting ordinary air compressors.

Pneumatically Operated Controllers

When processes must be controlled more accurately than is possible with a simple thermostat or pressurestat, some means of varying the power input with the load must be adopted, and compressed air, because of its flexibility, is usually employed for operating instruments of this type. The pneumatic controller is usually combined with a recording instrument, so enabling the variations of the controlled value to be recorded on a clockwork-driven chart, and at the same time operate the instrument. This makes it possible to check the accuracy of the control and make adjustments, the need for which may only become apparent after a period of use. The controller mechanism is designed to allow for the inevitable timelags which occur in its response to changes in the process and the taking effect of the correcting action. Both the amount and rate of the variation are taken into account so as to prevent over-correction and hunting. The controller is connected by pipes to the

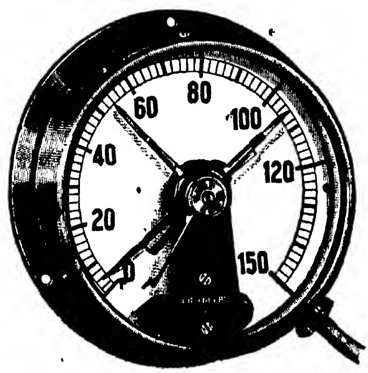


FIG. 21.—PRESSURESTAT
(*Budenberg Gauge Co., Ltd.*)

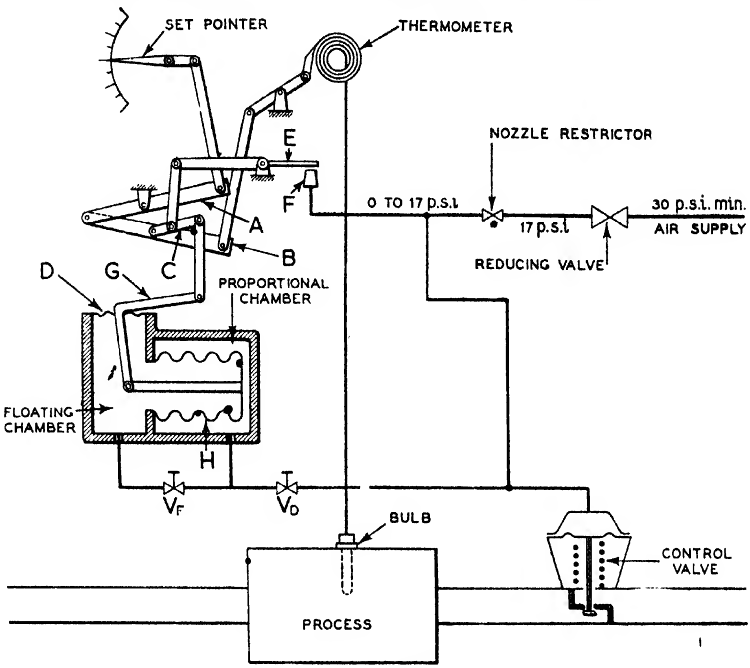


FIG. 22.—MECHANISM OF FULLY AUTOMATIC CONTROLLER (*George Kent, Ltd.*)

actuating elements which are usually pneumatic diaphragm valve or power cylinders, designed so that they open or close in proportion to the air pressure from the controller. Valves are used for controlling steam, gas, water, oil, etc. Power cylinders are employed when valves are unsuitable.

Fig. 22 is a simplified diagram of a fully automatic controller. The instrument mechanism incorporates two members *A* and *B*. *A* is pivoted to the main frame, and is moved up and down by the set pointer. *B* is pivoted to *A* and attached to the sensitive element, e.g. a thermometer. *B* has a link *C* which actuates the nozzle flapper, and is connected at the other end to the cranked rod *G*, described later. Compressed air passes to the nozzle through the restriction, and a small movement of the flapper (about 0·001 in.) can vary the pressure between restriction and nozzle from 0 to 17 p.s.i. These variations control the openings of the spring-loaded diaphragm valve. (In practice, an air relay is used between nozzle and valve to speed up the response.) If the temperature is too high, member *B* drops as shown, and lifts the flapper so that the end is uncovered and the internal pressure drops, partially closing the valve. The pressure is also reduced in the proportional chamber so that the bellows *H* expand and cranked

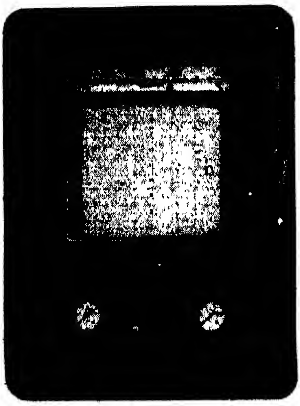


FIG. 23.—AUTOMATIC POTENTIOMETER CONTROLLER (*George Kent, Ltd.*)

rod *G* turns anticlockwise, so closing the flapper. This gives proportional control (Fig. 18 *B*). If air is allowed to leak through the adjustable restrictor V*f*, the pressure either side of the bellows is gradually equalised and tends to return to its original position, and proportional plus floating control is obtained (Fig. 18 *C*). The addition of a valve V*d* to slow up the transmission of pressure to the proportional chamber causes a delay which varies with the rate of change of temperature and so gives control as in Fig. 18 *D*. The above describes what happens when one isolated disturbance takes place. In practice changes are always taking place and the mechanism moves continuously, and the controls are set to give the best results for the particular process.

Applications

These controllers can be applied to temperature, pressure, humidity, flow, density, liquid level, *p*H, etc., and their uses include the control of temperatures of superheated steam, general and oil industry process plant, steel furnaces, kilns, etc. It is interesting to note that pneumatic control is used even when the temperature is measured electrically by a recording potentiometer (Fig. 23). However good a controller, it will not function satisfactorily unless it is possible to measure accurately and reliably the controlled value, and this must be kept in mind when designing a plant with which it is intended to use one of these controllers. It is also essential that the available supply of energy should be ample to meet any demands.

INSTALLATION AND MAINTENANCE OF INSTRUMENTS

If instruments are to be relied upon, it is essential that they should be properly maintained, as, if inaccurate, they may give a false sense of security with possibly disastrous results. Care in selection will be amply repaid, and if any doubt arises, it will be found that most reputable manufacturers are more than willing to offer advice and technical assistance. Installation is also important, and to-day the tendency is to group instruments in convenient positions where they can most easily be read by the users. Panel mounting usually involves pipe-lines, and these must be run so that they can be bled to remove air locks if liquid filled, or condensation if working on gas or air. Steam should always be condensed by a siphon or chamber, so that connecting pipes are water filled. Pressure-operated instruments are affected by the head of liquid in the connect-

ing pipe, and if this amounts to more than 1 per cent. of the working pressure, allowance should be made during calibration. The capillaries of distance thermometers need installing with care as, if damaged, the instrument becomes useless. They should be laid in such a way that there would be no difficulty in subsequently removing the instrument should this become necessary.

Instrument Department

If a large number of instruments are in use, it is desirable to have a separate instrument department, which should preferably be self-contained, well lighted, and centrally situated. The minimum equipment would include a bench with vices, drilling machine, and, for making spare parts in an emergency, a small bench lathe. There should be storage racks for spare instruments and those awaiting repair, and a clear wall for suspending the larger types whilst testing. The testing equipment itself would normally include an electrical test panel, with tappings for all voltages likely to be needed, indicator lamps for circuit testing, and a Grade 1 ammeter and voltmeter. If many thermometers are installed, then a gas or electrically heated test bath is necessary. This should be filled with glycol for temperatures up to 150° C. and a low melting-point fusible salt for higher temperatures. Oil can be used, but is not recommended, as it is very sluggish at low temperatures and fumes badly at the higher ones. For checking temperatures up to 400° C. etched-on-glass thermometers with an N.P.L. certificate are useful. If there are many thermocouple pyrometers, then a potentiometer-type resistance box is recommended, as it can be used to measure temperature with standard thermocouples and also for checking indicators. A deadweight tester is to be preferred for pressure-gauge testing, as this gives a permanent standard. Pressure gauges graduated to less than 20 p.s.i. and vacuum gauges are best tested against a liquid column (Fig. 2).

It is always best to service instruments regularly, and a card index should be kept for each instrument to make sure it is overhauled at regular intervals.

J. R. F.

ELECTRONICS IN MECHANICAL ENGINEERING

MANY problems which are difficult or impossible to solve by ordinary means can often be done efficiently by electronic methods. Electronically produced high-frequency currents, with characteristics very different from the standard 50-cycle currents, can be used for heating and in the operation of instruments. Electronic switches can make thousands of operations a minute, and are used, amongst other things, for operating variable-speed D.C. motors from the A.C. mains.

Photocells and proximity sensitive circuits with electronic amplification operate without physical contact, and will control any power with a speed and sensitivity unapproachable by other means.

Diode Valve

The heart of every electronic device is an electron tube, many types of which have been developed from the radio valve. The simplest of these is the diode valve, the principles of which are illustrated in Fig. 1. As long as the anode is kept at a positive potential in respect to the cathode, the negatively charged electrons emitted by the hot cathode will be attracted by the anode and a current will flow. The flow ceases as soon as the anode potential falls or becomes negative. If an alternating current is applied, it will be rectified to D.C. as shown. By using two valves (or two elements in one envelope), both halves of the A.C. wave can be used.

The smaller sizes have a metal filament, but this is a weak feature of all valves. In the mercury bulb rectifier it is replaced by a pool of mercury, which, after an initial heating, provides a constant source of electrons that will carry very large currents. The mercury rectifier has the further advantage that one bulb will rectify 3-phase current, whereas three or even six filament-type valves are needed for the same purpose.

Triode Valve

For more general use the valve has a grid (Fig. 2) interposed between anode and cathode, which makes it possible to control the main current from anode to cathode by applying small voltage variations to the grid. In the example, the grid is controlled by the minute variations in the photocell as the light intensity changes, and this causes the valve to pass sufficient current to operate an electromagnetic relay. The thyratron is a modification of the triode valve. It commences conducting when both grid and anode are positive, but only ceases when

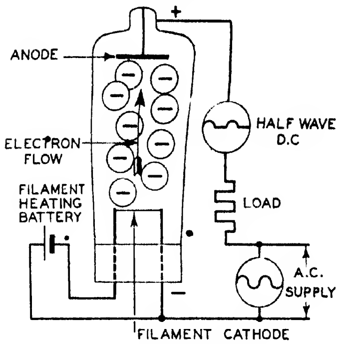


FIG. 1.—DIODE VALVE

the outer circuit is broken or its polarity is reversed.

High-frequency Eddy-current Heating (Metals)

An alternating current of sufficiently high frequency (1 Mc.) can be used to heat metals by placing them within or near a coil of copper tube connected to a generator. With a valve oscillator, powers up to 25 kW. can be generated, which is ample for most workshop needs. The eddy currents induced in the metal cause it to become heated and, as the power can be concentrated, the time taken is usually much shorter than with other methods.

By suitably shaping the coil, only the relevant part of the body need be heated, and by manipulation heating can be confined to the surface of the metal only. Fig. 3 shows the applicator unit and examples of work done on a 2-kW. eddy-current heater.

The applicator unit to which the coil is plugged is shown at (*A*); soft soldering a Bourdon gauge tube (10 sec.) (*B*); hardening the ends of a tappet (*C*); brazing the end of a tappet (*D*); soft soldering fuse caps on a conveyor (3,000 per hour) (*E*); annealing brass tubing by passing it through the coil (1 ft. per minute) (*F*).

Eddy-current heating is most suitable for repetition work, as a special coil must be made for each part and the best shape determined by trial. When large numbers of similar parts have to be heated, it is often possible to run them past the coil on a conveyor, whilst for parts which are heated individually, a timer cuts off the current automatically. By fitting high-frequency coils in cavities in moulds for plastics, it is possible to heat them very evenly and efficiently.

Soldering and Brazing

Parts of any size or shape weighing not more than a few pounds can be heated in a few seconds just where the joint is to be made, whereas with a flame it would probably be necessary to heat the whole part. Little or no scale is formed, and the exact amount of solder can be applied in wire form.

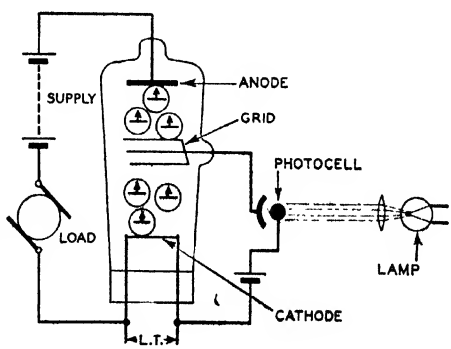


FIG. 2.—TRIODE VALVE

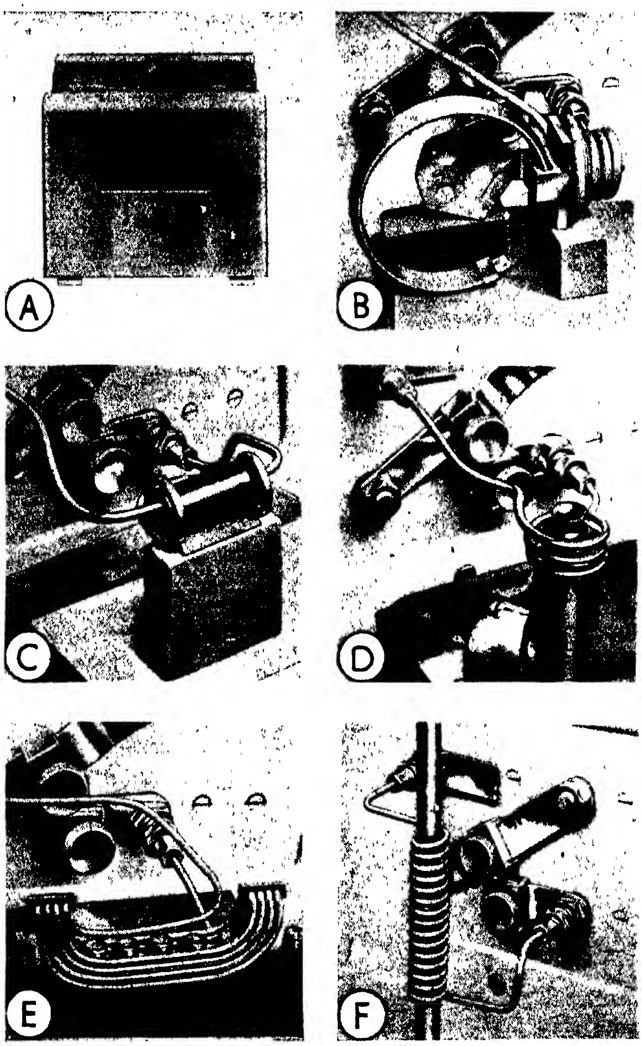


FIG. 3.—EDDY-CURRENT HEATING

A. Applicator unit.
B. Soft soldering a Bourdon gauge.
C. Hardening ends of a tappet.
D. Brazing the end of a tappet.
E. Soft soldering fuse caps on conveyor.
F. Annealing brass tubing.

(*Delapena & Son, Ltd.*)

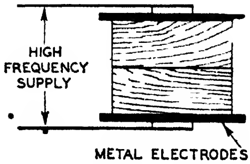


FIG. 4.—DIELECTRIC HEATING
The two metal plates are connected to the high-voltage H.F. generator, and between them is placed the material to be heated, which therefore becomes the dielectric of a condenser.

Hardening, Tempering, and Annealing

Eddy-current heating is particularly suitable for local and case-hardening small parts made from steel with more than 0·5 per cent. carbon and is used for pins, gear teeth, rockers, and other small parts. With a high-power generator, heating is so rapid that it does not have time to penetrate to the centre of the work, and an adequate hard case can be obtained by heating for 5 seconds and then quenching. As the centre remains cold there is almost no distortion, and grinding operations may often be eliminated. Because of the very local heating it is possible to anneal small portions of hardened parts, such as the ends of chain pins, where they are riveted.

High-frequency Dielectric Heating (Non-metals)

The application of a high-frequency current to two metal plates (Fig. 4), separated by a non-conductor (dielectric), causes molecular disturbances in the dielectric which generate heat within the material, the rise in temperature being almost uniform throughout. On the other hand, with convection or conduction heating methods, penetration of heat to the centre is very slow, and it is almost impossible to obtain uniform heating in a reasonable time.

Using a high voltage and high-frequency (up to 80 Mc/s) materials, such as thermo-setting moulding powders (Bakelite), can be heated before placing in the mould, so that curing starts immediately, instead of having to wait for heat to be gradually conducted from the hot mould. Not only is there a considerable saving in time, but it is found that the quality of moulding is greatly improved; in fact, some mouldings can only be made satisfactorily if the powder is preheated in this way. Dielectric heating can also be used for making laminated materials and for the evaporation of moisture from substances which are difficult to oven dry.

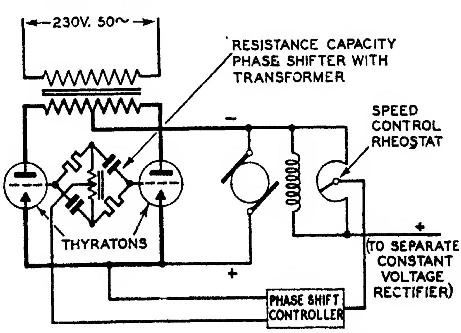


FIG. 5.—OPERATING A VARIABLE-SPEED D.C. MOTOR FROM THE A.C. MAINS

Motor Speed Control

The problem of providing stepless variable speed control on machine

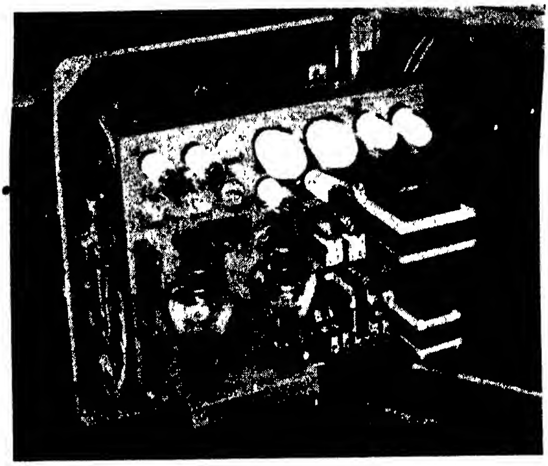


FIG. 6.—"EMOTROL" ELECTRONIC MOTOR CONTROLLER

The controller panel, comprising thyratrons, transformers, contactors, etc., is mounted on the door let into the base of a machine tool. The stop-and-start buttons and variable-speed control are mounted above.

(*The British Thomson-Houston Co., Ltd.*)

tools and other machinery has been largely solved by the electronic rectification of mains A.C. to variable voltage D.C. This makes it possible to use variable-speed D.C. motors of conventional design without the A.C. motor and D.C. generator associated with the Ward-Leonard system.

A speed range of up to 100 : 1 can be attained on motors up to 1,000 h.p. Normally, the speed is adjusted by a control knob, but if necessary, speed changes can be made automatically by a timing device or a trip on the machine. Where groups of motors are used on continuous processes, it is possible to run each motor at a different speed to allow for material stretch or other variables. Motors driving reeling drums can be slowed down as the amount of material on the drum increases, so as to maintain constant linear speed.

The principles of operating a variable-speed D.C. motor from the A.C. mains are illustrated in Fig. 5. The D.C. supply is furnished by a bi-phase grid-controlled thyratron rectifier fed from a transformer. The voltage supplied to the motor armature by the thyratrons is varied by altering the point in the A.C. cycle when the grid potential is sufficiently positive to cause the thyratrons to "fire" (commence conducting). This is done by supplying to the thyratron grid an alternating potential lagging 90° behind the main supply, through a small transformer and resistance/capacity phase shifter, and superimposing on this, through the phase shift controller, a steady potential that can be varied.

FIG. 7.—PROXIMITY METER

(*Fielden Electronics, Ltd.*)

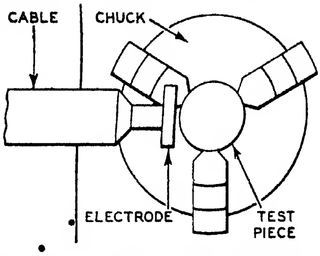


FIG. 8.—CHECKING LATHE BEARINGS WITH A PROXIMITY METER

The hand-operated rheostat controls the point in the cycle at which the grid potential becomes positive and, therefore, the firing point. If this is early, the thyratrons pass the full voltage, whilst if late, only a proportional voltage is passed. In the example the motor field is energised from a separate constant-voltage transformer, but for wide speed ranges the field voltage must also be varied. For the sake of clarity, details are omitted of the contacts, automatic compensation, etc.

Up to 45-h.p. thyratrons (Fig. 6) can be used for giving the variable-voltage D.C. supply; two thyratrons, operating from single-phase A.C. being used up to 3 h.p., and three thyratrons operating from 3-phase A.C. for the larger motors. The mercury-bulb rectifier with grid control can also be used for the supply of variable-voltage D.C., and it has the advantage that only one bulb is required.

Variable-speed D.C. motors usually give a constant torque, so that the horsepower varies with the speed. On low speeds the power available is thus considerably less than that from a gearbox driven by a constant-speed motor, and this point should be noted when assessing the size of motor.

Instruments and Measuring Devices

The application of electronics to instruments enables an enormous increase of sensitivity to be obtained, whilst the absence of any inertia makes it much easier to follow quickly changing phenomena. Generally, operations can be carried out without physical contact, either by using a photocell or by altering the capacity of a high-frequency circuit.

Proximity Meter

An example of an instrument using the latter method is the proximity meter, which can be adapted for measuring very small movements, such as those caused by loading a machine or structure, or by the forces on the slides of a machine tool whilst cutting. It can be used for measuring the sizes of metal parts, and the gauging of quantity-produced parts with automatic rejection of faulty ones is also possible. Sheets and foils can be gauged continuously during

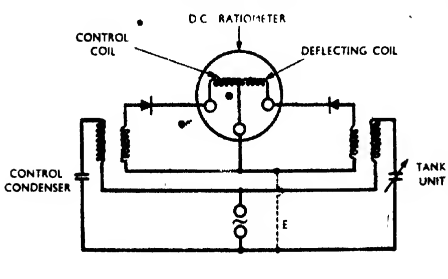


FIG. 9.—"PACITOR" ELECTRONIC TANK GAUGE

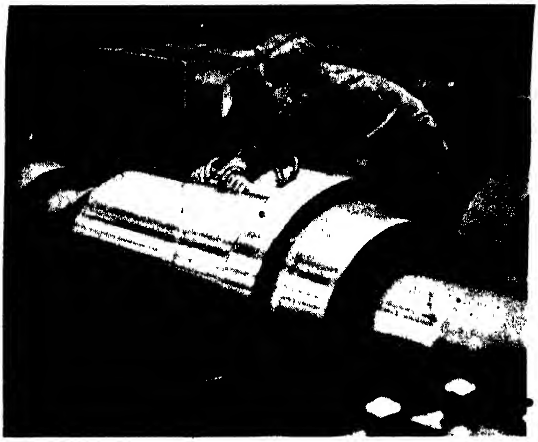


FIG. 10.—SUPERSONIC FLAW DETECTOR
(*English Steel Corporation, Ltd.*)

the manufacture without fear of damage, as there is no need to make contact.

A proximity meter is illustrated in Fig. 7. A co-axial cable carries an electrode which is connected to an oscillator and amplifier, changes of the electrode capacity of as little as 0·001 mmf. being registered on the meter or used to operate a switch. The capacity of the electrode is affected by the movement or approach of a solid body, a liquid, or a flame. It is also affected by variations in dimensions or composition of a body.

Fig. 8 illustrates an application of this instrument. To check lathe bearings with a proximity meter, a test piece is turned and left in position in the chuck.

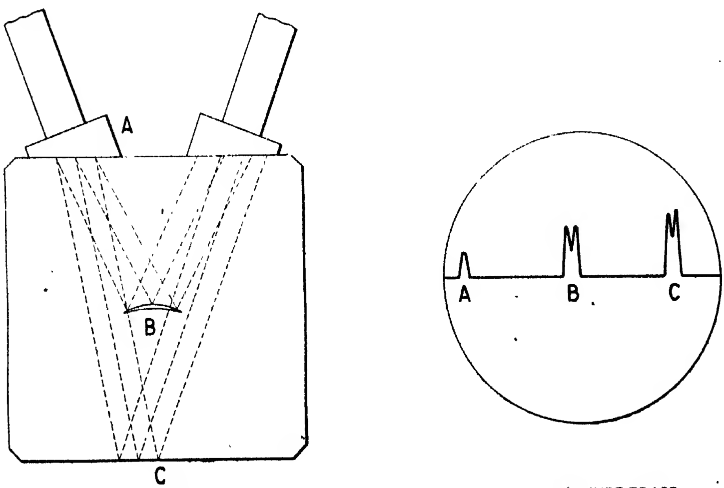


FIG. 11.—THE PRINCIPLE OF THE SUPERSONIC FLAW DETECTOR

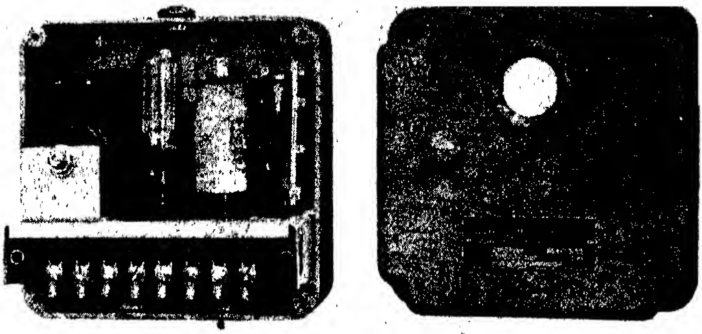


FIG. 12.—PHOTOCELL SWITCH

Photocell (*centre*), amplifying valve (*right*), and magnetic relay (*left*). The unit is housed in a substantial case with a small window. (*The British Thomson-Houston Co., Ltd.*)

The proximity meter electrode is mounted on the cross slide and advanced within about 0·005 in. of the work. The meter is set to zero, and the slide is then advanced 0·001 in. and the sensitivity adjustment operated until the meter gives a full scale deflection for the 0·001 in., enabling readings to be taken of any out of the roundness to 0·00001 in.

Electronic Tank Gauge

Many electrical controlling instruments now incorporate an electronic switch which is operated by a vane on the instrument pointer. The vane acts as a coupling to two coils forming part of a valve system, a movement of 0·001 in. of the vane altering the coupling sufficiently to stop or start the flow of electrons in the valve and so provide power for operating the main circuit.

The variable-condenser principle forms the basis of the Simmonds electronic tank gauge which has a number of advantages over pressure- and float-operated gauges, especially when the tanks are of irregular shape, such as those fitted to aircraft. The condenser consists of two concentric Duralumin tubes attached to a suitable flange and fixed in the tank in a vertical position. For fixed tanks of regular shape one unit is sufficient, but for aircraft tanks it is necessary to use several units connected in parallel to counteract the effects of changes in altitude of the aircraft.

As the dielectric constant of petrol is twice that of air, the condenser passes twice as much current when the tank is full as when it is empty, and this is indicated on the two-coil ratiometer dial which is calibrated to suit the installation.

The wiring of a "Pacitor" electronic tank gauge is shown in Fig. 9. This circuit compares the capacity of the tank unit with that of the control condenser,

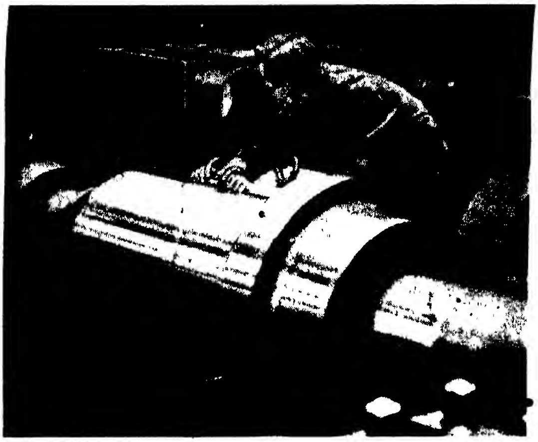


FIG. 10.—SUPERSONIC FLAW DETECTOR
(*English Steel Corporation, Ltd.*)

the manufacture without fear of damage, as there is no need to make contact.

A proximity meter is illustrated in Fig. 7. A co-axial cable carries an electrode which is connected to an oscillator and amplifier, changes of the electrode capacity of as little as 0·001 mmf. being registered on the meter or used to operate a switch. The capacity of the electrode is affected by the movement or approach of a solid body, a liquid, or a flame. It is also affected by variations in dimensions or composition of a body.

Fig. 8 illustrates an application of this instrument. To check lathe bearings with a proximity meter, a test piece is turned and left in position in the chuck.

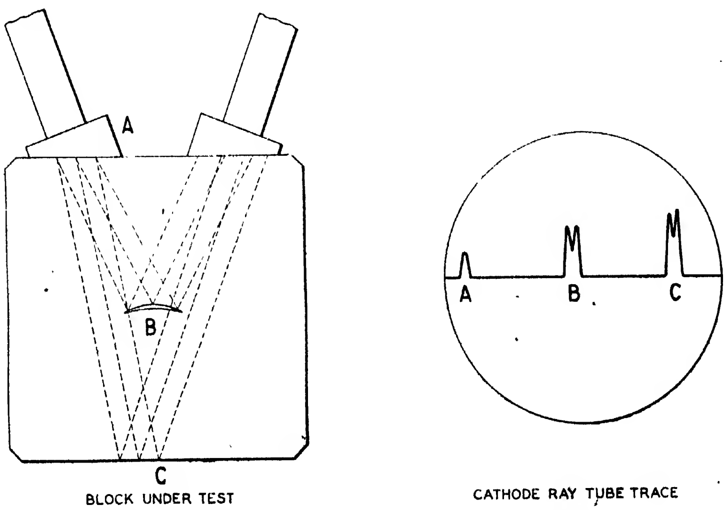


FIG. 11.—THE PRINCIPLE OF THE SUPERSONIC FLAW DETECTOR

this method can be employed for detecting flaws in metals and for measuring the thickness of parts when only one side is accessible.

The Hughes supersonic tester can deal with thicknesses from about ½ in. to 30 ft., although some difficulty is experienced with the thinner specimens. It is, however, very convenient for making non-destructive tests on large bars, ingots, forgings, and welded parts. It is possible to explore the whole length of a turbine rotor or propeller shaft, and in steel works the cutting of the discard from billets can be done with far less waste than formerly. The method requires an experienced operator, and it may be necessary to compare the results given by the tester with some sectioned test pieces, before it can be said with certainty which parts are bad and which are sufficiently sound.

In Fig. 10 the tester is being used for examining a large steel forging during machining. The probes held by the operator are connected to a power unit (not shown in the illustration). The results are shown on the cathode-ray tube.

Fig. 11 illustrates the principles of the supersonic flaw detector. Energy pulses produced by the power unit are converted by the quartz oscillator (*A*) into supersonic waves which travel in the material until they meet a boundary, when they are reflected and picked up by the probe (*B*). The echoes are shown as pips on the cathode-ray trace, the distance of the pips from the transmission mark (*A*) being in proportion to the depth of the reflecting surface. The fault (*B*) is shown by the flaw mark (*B*) on the trace and (*C*) is the bottom mark.

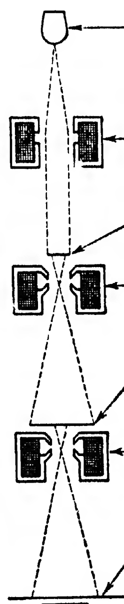


FIG. 15.—PRINCIPLE OF THE ELECTRON MICROSCOPE

Photocells

These are some of the most useful of the electronic devices, and as they form the basis of film-sound reproduction, much time has been spent on their development. Their use in engineering has been retarded by the comparatively high price of the commercial photocell actuated relays and the lack of appreciation of their possibilities.

The response of a photocell to a change in light intensity takes about a hundred-millionth of a second, which is several million times as fast as a man can respond to a visual impulse, even if no allowance is made for the human fatigue factor. The standard photocell relay will operate up to 300 times a minute, but for special work much faster speeds are possible.

A commercial photoswitch (Fig. 12) consists of the photocell, a valve amplifier, and a relay, the latter usually being of the electromagnetic type. The cell can be operated by changing from darkness to light or vice versa, or by varying the light intensity.

If a light is shone on the cell from a lamp with projector lens, and this is interrupted by a solid body passing across it, the switch can be used for counting objects on conveyors, opening and shutting factory doors at the approach of trucks, operating safety devices whilst a person or hand is in the danger zone, or stopping feeds when a hopper is full. Sometimes, for instance, the light is masked whilst things are normal and unmasked if a strip of material should break. If the light is attached to a moving part, it can be used to operate the switch when the part reaches some predetermined position, as in the aligning of swing bridges where mechanical locks are unsatisfactory.

Some photocells are particularly sensitive to red light, and can be employed for operating mechanisms dealing with red-hot sections in rolling mills. Some of the most interesting applications are those where the cell responds to light reflected from a surface. By this means it is possible to sort things by colour as fast as they can be handled by an automatic feeding mechanism.

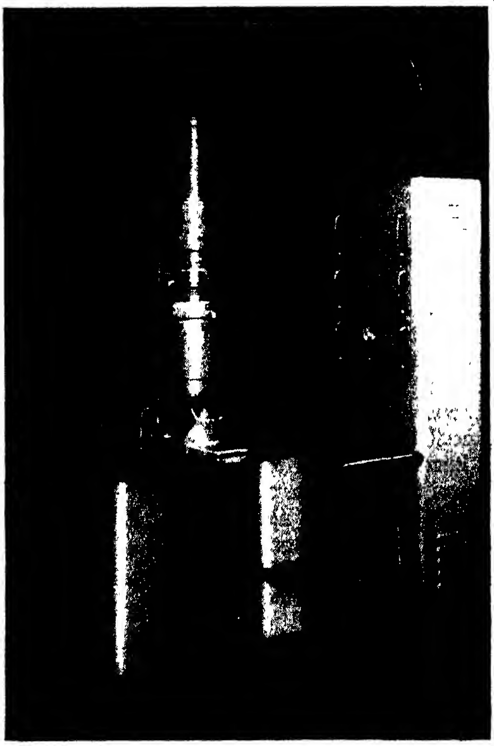


FIG. 16.—ELECTRON MICROSCOPE
(*Metropolitan Vickers Electrical Co., Ltd.*)

Photocell Tracing Head

A recent development is its use for guiding a tracer so that it follows the outline of a drawing automatically. By connecting the tracing head to a pantograph mechanism, it is possible to cut out irregular shapes accurately and automatically, or to operate an engraving machine.

A tracing head (Fig. 13) connected to the burner carries a lamp, photocell, and tracing wheel. The wheel traverses the head over the drawing, being rotated continuously by one electric motor and steered by another. To prevent damage to the drawing the wheel does not rest on it directly, but on a transverse cylinder, which it rotates and which is geared to the supporting wheels. A pinpoint of

FIG. 17.—GAMMA-RAY RADIOGRAPHY (*Johnson Matthey & Co., Ltd.*)

light is projected on the drawing so that it is half on the black line and half on the white background, in which position the steering motor is stationary. If the spot deviates from this position the intensity of the light picked up by the photocell varies, and this is arranged to operate the steering motor in the direction needed to bring the spot back. To prevent interference from the ordinary lighting, the photocell is made to operate only from the special light from the lamp.

The Cathode-ray Oscillograph

The cathode-ray tube enables the engineer to explore the behaviour of moving machinery with a precision which would otherwise be impossible. Regularly recurring phenomena are depicted as a persistent curve which can be examined by eye, whilst transient shocks are photographed with a high-speed camera. By using one or other of these methods, almost any type of stress or strain can be measured or analysed.

The cathode-ray tube is diagrammatically illustrated in Fig. 14. An electron beam is directed on to the fluorescent screen so that the path of the beam can be observed. Two pairs of plates or electro-magnets are arranged as shown which deflect the beam according to the electrical impulses they receive. The *XX*

plates deflect the beam horizontally, and are usually connected to a timing device which moves the beam from L to R at an adjustable frequency. The return from R to L is made instantaneously and cannot be seen. The timer can also be operated from an engine camshaft. The YY plates deflect the beam vertically, and are connected to a source of potential or current which varies with the quantity to be measured. When measuring a persistent phenomenon such as vibration, engine cylinder pressure, etc., the same pattern is repeated regularly and can be seen by eye. Isolated impulses cannot be observed directly, and must be photographed with a high-speed camera.

The cathode-ray tube also makes a convenient dead-beat galvanometer for taking a number of readings on strain gauges, as no time is lost waiting for the instrument to come to rest.

For pressure analysis, electrical pressure gauges, having either a carbon block whose resistance varies with pressure or a diaphragm which changes the capacity of the condenser, are used. Vibration is also measured by the condenser effect, one plate of the condenser being fixed to a solid support and the other attached to the part under test.

Strains are measured with a wire resistance strain gauge, consisting of a flat element which is glued to the surface of the part under test, any movement causing a change in the resistance of the wire. It is, of course, necessary to employ the appropriate electrical circuits between the measuring devices and the oscillograph.

The Electron Microscope

The magnifying power of an optical microscope is limited by the wavelength of light, but by using electrons instead it is possible to obtain considerably greater magnification with a present-day limit of 100,000 times. By this means, details of the structures of metals and alloys can be studied which would otherwise be impossible. The specimens themselves are extremely small and must be thin enough for the electrons to penetrate. If this is not possible, then a replica of the surface has to be made in a transparent material.

The specimens are first mounted on a collodion film, and this is supported on a $\frac{1}{8}$-in. diameter grid made from 200-mesh copper gauze. The electrons are produced by a hot cathode gun and accelerated with 25–100 kV. The beam is focused on to the specimen by the condenser lens, and the image is enlarged by passing through successive magnifying lenses (Fig. 15). For the highest magnifications an additional lens is used. The lenses are iron-shrouded solenoids which are focused by varying the excitation current. The image falls on to a fluorescent screen, when it is visible to the eye, or on to a photographic plate.

The electron microscope gives a great depth of focus, and very clear pictures are obtained which can be enlarged for greater magnification. Because electrons are absorbed by an atmosphere, the microscope must be completely evacuated before use, but vacuum pumps can do this in three minutes.

In the electron microscope shown in Fig. 16, the microscope proper is mounted on a control desk, so that the fluorescent screen is at a convenient height

for viewing through the eyepiece. The plate holder is immediately below. The power unit and vacuum pumps are separate.

Radiography

The non-destructive testings of castings, weldings, etc., can be done by taking shadowgraphs with the aid of X-rays and gamma rays. The records are usually made on photographic film, and a lengthy exposure may be necessary because of the opacity of the subjects. Examination of the radiographs by an experienced person makes it possible to detect flaws which would be detrimental in service. X-ray outfits are similar to those used for other work, but high powers are needed for thick parts.

Tubes working at one million volts are employed for large forgings and are able to give a radiograph through 4½ in. of steel in a few minutes. So large a power is unusual, and X-ray tubes working on 300 kW. or less are more common.

For small parts, radium-emitted gamma rays are often more convenient now that it is possible to hire radium from the Government Radio-chemical Centre at Amersham.

The radium is supplied in special protective carriers, and the usual practice is to stand the parts in a circle with the radium container at the centre (Fig. 17), each part having a film carrier behind it. The radium is extracted from the container and replaced by an automatic lifter operated by a time switch, so making it unnecessary for anyone to come in contact with the gamma rays.

J. R. F.

INSTALLATION OF OIL AND PETROL ENGINES

THIS description is intended to cover all engines of the high-compression, cold-starting, heavy-oil type, both mechanical and air injection.

The tendency for most land installations, such as private generating stations, pumping sets, and direct mill or factory drive, is towards the adoption of the airless or mechanical fuel-injection type. This is primarily due to initial cost, greater mechanical simplicity and, therefore, increased reliability, and the ability to use a wider range of low-grade fuel oils.

The purpose of this article will, however, be served by assuming that the remarks apply equally for both types, since the general construction and methods of assembly will be the same.

Vertical or Horizontal Engines?

The horizontal class of engine is still very popular and generally employs the mechanical system of fuel injection. For layouts embodying a drive from engine to mill shaft, either direct or indirect, this class of engine is favoured, owing to the lower head room required and the slower engine speed which generally enables a satisfactory drive to be obtained.

The vertical engine is eminently suitable for coupling direct on to electric generators, compressors, pumps, for which it has many special advantages.

THE ENGINE-ROOM

The buildings or engine-rooms for oil-engine installations will depend naturally on the size of plant being installed and the amount of capital available for this item. Generally speaking, money spent on a substantial, dry, and clean engine-house will not be wasted, and will result in more efficient and economical running, and longer life from the engine. No hard-and-fast rule can be laid down on the matter of buildings, for all considerations will obviously be governed by the degree of permanency required and the factors mentioned previously.

Modifying an Existing Structure

For waterworks and similar installations, stone and brick-built structures are usual, whereas in some cases it might be found desirable to use an existing building to house the plant.

In this case it may be necessary to modify or improve the old structure to suit its new purpose. For instance, buttresses may have to be added to carry crane supports, windows may have to be increased or enlarged if the house is

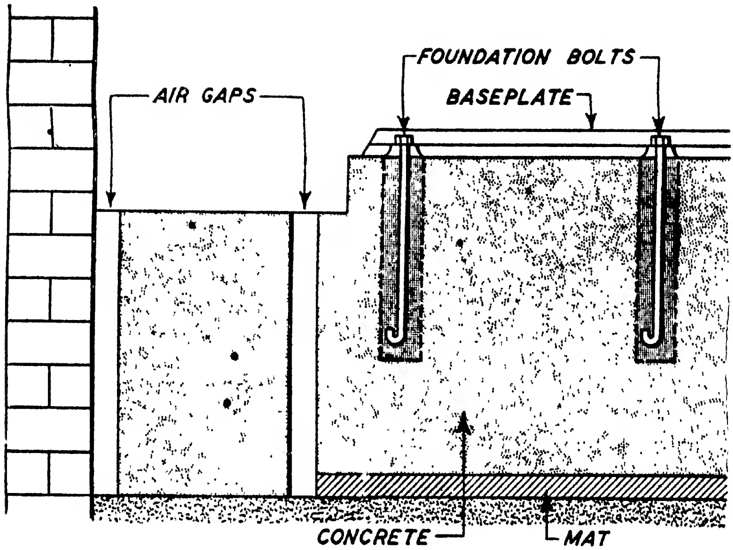


FIG. 1.—SPECIAL ANTI-VIBRATION ENGINE BED

poorly ventilated or lighted, and doors and roof may have to be improved to exclude dust and damp. A good floor may also have to be laid which should be free from anything that is likely to make dust or grit in the course of time.

Heating the Building

Large plants, and those including electrical gear, which are in particularly cold or damp situations, or which are likely to be shut down for any length of time, may require heating arrangements to maintain a reasonable temperature and to keep the place dry These usually take the form of a hot-water boiler and circulating pipes or radiators, and the boiler is, as a rule, either coal, coke, or oil fired. If the latter, it may, if necessary, be placed in the engine-room, but if coal or coke fired, it is almost imperative that the boiler shall be in a separate compartment on account of the dust and dirt entailed.

A Satisfactory Building

From the foregoing it might seem that elaborate engine-houses are essential for oil-engine plant, and although a good building is desirable and should be provided where possible, it must be mentioned that extremely good results have been and are being obtained from engines with a very meagre covering. A cheap and quite satisfactory building for average use consists of a steel-frame skeleton with galvanised corrugated-iron covering.

If financial considerations will allow, it is an advantage to line the inside with asbestos sheeting or with suitable light boarding, as this will assist in preventing extreme cold and the ingress of damp and moisture during bad weather.

Arranging for Lifting Tackle

The four main stanchions of the building can be arranged to carry the crane rails. If there is only one engine, however, and parts to be lifted do not exceed, say, 15 cwt., it is often a simple matter to support a girder along the centre of the room and arrange lifting tackle from this. Overhauling is sometimes provided for in this way when the plant comprises more than one engine, but in these cases it is not much more expensive to fit a crane of the hand-travelling type, which will be generally more satisfactory in use. If tackle is rigged from a single girder, two men are usually required for the removal of pistons, etc., whereas a crane can be operated with ease by one mechanic. This point is worthy of consideration when deciding on the initial outlay of such an accessory.

Consult the Engine-maker

Reputable firms are always pleased to give their views as to the best type of building for any particular installation in hand, and will usually supply drawings showing the best layout to adopt, and advise generally on the problems of any individual installation. It may be found that the local builder or architect sees fit to alter the engine-maker's arrangement drawing, and if this is the case, these modifications should be communicated to the engine-maker so that piping, etc., can be modified accordingly where necessary. The internal dimensions of the engine-room, i.e. length, breadth, and head room, should, however, be adhered to, also the position of doors and windows, which are of similar importance.

LAYING THE FOUNDATIONS

The foundations for the plant should be put in carefully to the maker's special foundation drawing.

If the person installing the engine has not previous knowledge of local conditions affecting foundations, it is as well either to get advice on this point, or else to make the engine-builders fully responsible for the adequacy of the foundations shown on their drawings.

The work of laying the foundations is usually easier if the building is first completed.

Installing Overhead Tanks

Before the builder removes his scaffolding, his services should be enlisted to place in position any overhead water or fuel tanks. It may be found an advantage to have these fixed before the roof is finished, and this work should then form part of the builder's contract, and delivery of the tanks arranged accordingly.

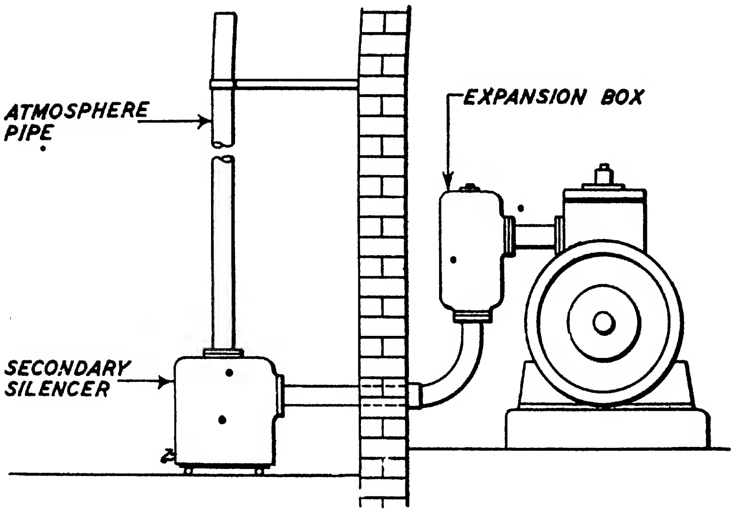


FIG. 2.—EXHAUST SYSTEM WITH SECONDARY SILENCER

Concrete Work to do at the Same Time

In proceeding with the work of excavating the foundations, it should be remembered that concrete supports for fuel and water tanks, exhaust pits, and pipe trenches should be made at the same time. It may be that the men doing this part of the work have had previous experience of such work, but in any case the makers' notes and instructions, contained either in book form or on the drawings, should be followed carefully. These notes contain the knowledge of long experience, and if ignored an unnecessary waste of time and expense may easily result.

Excavating the Ground

The ground should be taken out as indicated in the drawings, and the sides of the excavation timbered if necessary in soft ground. If the subsoil is not solid at the depth indicated on the drawing for the foundation in question, the depth must be increased until a good firm bottom is obtained. In the case of water being encountered, pumping and close boarding may have to be resorted to.

Making and Placing Wood Boxes for Concrete

Make an open box representing the engine bed or pedestal. Check the centre lines to the drawing. Now fix the box or template in the correct position relative to the building or other machinery. This can be done by nailing it to two long planks resting on pegs or on the ground. The underneath side of these pieces of wood should represent the top of the foundation proper.

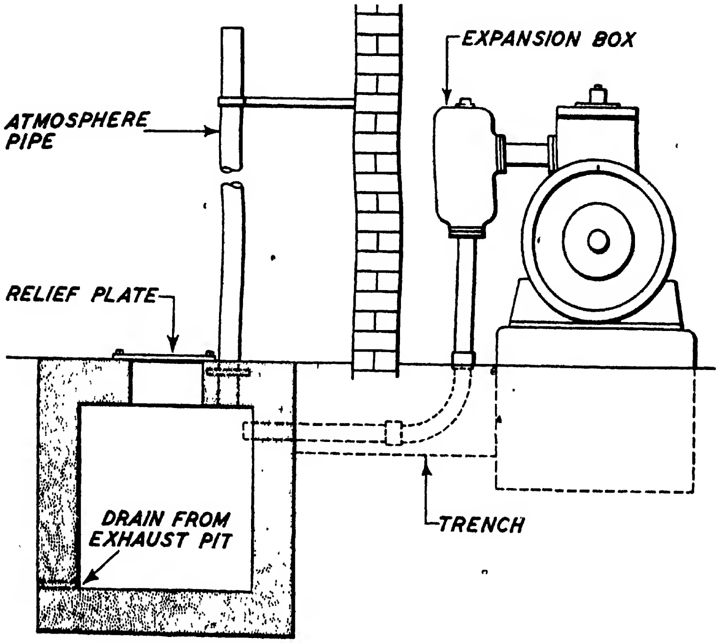


FIG. 3.—EXHAUST SYSTEM WITH PIT SILENCER

If the engine is to be direct-coupled to a pump or electric generator, the foundations for the complete set should be cast in one piece. In any case, the outer bearing pedestal and such projections as the air-compressor or water-pump blocks must be cast at the same time as the rest. The boxes for these pedestals are also supported at floor-level in the same way as the engine bed.

Providing for Bolt Holes in Concrete

Holes in the concrete for foundation bolts should extend below the bottom of the bolts. The boxes for these holes should be made of about ½-in. wood lightly nailed together, so as to be easily collapsible after the concrete has set. Wooden blocks must on no account be used. It is a good plan to move the boxes slightly before the concrete has thoroughly set so as to give a small clearance.

Check the Position of Boxes before Concreting

Before commencing the actual concreting work, the positions of the various boxes should be carefully checked to the drawing and existing plant, and care should be taken to see that there is available sufficient material to complete the work in one continuous operation.

Mixing the Concrete

The concrete may now be mixed, and should consist of roughly four parts of good clean ballast or equal, not larger than 1½ in., two parts of sharp sand, and one part of best Portland cement. Quick-setting cement may be used if desired, but should be placed in position with as little delay as possible.

Filling in the Concrete

The whole of the concrete should be filled in carefully. Take care to see that the boxes are not disturbed and that the work proceeds continuously, so that one solid homogeneous block of concrete is formed. The top surface should be left rough about ¾ in. below finished level for grouting up. Concreting should not be done in frosty weather.

Providing Clearances in Concrete

The foundations should be about 3 in. clear of wall footings. Flywheel races, clearances for pulleys, etc., and the recesses under the engine bed can be formed by light boxes placed in position as the foundation work proceeds.

Drainage for the flywheel race and the pipe trenches can be provided by earthenware pipes fixed in the concrete.

Kerbing for the trench covers should be grouted in and care taken to see that the top is flush with the finished floor-level.

ERECTION OF ENGINES

When the foundation work has been finished and all bricks, dust, and dirt, etc., such as are usually connected with a building job, have been removed and the concrete has had a few days to harden, erection may commence.

Keeping Engine Clean during Erection

At this stage it will, perhaps, be as well to give a few words on the subject of cleanliness. It is most important that any oil engine, or for that matter an engine of any sort, should be kept scrupulously clean whilst it is being erected, and care should be taken that no bearings or wearing parts go together with foreign matter between them. Given a clean engine, clean fuel oil, clean lubricating oil, and clean cooling water, sterling service can be expected from an oil engine, whereas dirt accumulated during erection may take years off the ultimate life of the engine.

Conveying Parts to the Site

It is usual for one door of the engine-house to be made sufficiently wide to allow for the machinery in its various parts to be either carried or rolled in. Where an extra wide doorway is provided, it may be possible to run the lorry close up to the foundation and slide such a part as the bedplate direct on to its base, thus saving a good deal of time. In some cases it may be advisable to leave part of the wall adjoining the doorway unfinished, so as to allow the lorry or trailer to enter.

Lowering Bedplate on to Foundation

If the lorry should happen to be higher than the foundation, timbers should be packed on the latter until the respective levels are the same, and the part can then be rolled on to the wood packing and gradually lowered by removing the packing piece by piece. Before the final packing is removed and the bed finally dropped on to the concrete, the wires which are fastened to the foundation or holding-down bolts should be drawn through the bolt holes in the bedplate.

Placing Bolts in Position

When all packing has been removed, the bolts may be drawn up through the bolt holes by means of the wires, and the nuts screwed on until about one thread short of a full nut. In this way, when the nuts are finally drawn down tight, the bolts will not project through the nuts.

Checking Position of Engine

The distance from the centre line of the crankshaft to the finished floor-level is now checked with the drawing, and the bedplate raised by means of steel wedges until this dimension is correct. At the same time dimensions from the centre line of the crankshaft to the wall, and from No. 1 cylinder to the wall, should be checked, likewise leading dimensions relative to other plant or machinery, and the position of the bedplate altered as may be necessary.

Belt or Chain-drive Alignment

Where the engine has a belt or chain drive to existing machinery, the crankshaft should be placed in its bearings with pulley, etc., fitted and a line taken to ensure that the belt will run true on both pulleys.

Checking Direct-coupling Alignment

For checking alignment relative to existing plant to which the engine is to be direct-coupled, the respective half-couplings should be fitted and the faces brought to coincide. Feeler gauges will show whether alignment is correct.

Connecting Shafts by Flexible Coupling

Fig. 4 shows a method of connecting two shafts together by means of a flexible coupling, and on most types of coupling there are machined faces intended for use by the erector, which greatly facilitate the work of lining up. There is a tendency for erectors to be somewhat careless over this operation and to rely on the flexibility of the coupling itself to prevent trouble. Whilst most good flexible couplings will compensate for any reasonable errors in alignment, the shafts should be set in line as shown, and should be adjusted in line as far as practicable, in order to avoid undue wear of the belt and possible bearing troubles.

The actual procedure to be followed in lining up a flexible coupling will vary with different types and makes, but the question of correct alignment will always be important, and a method of checking can easily be devised to suit the particular make of coupling in question.

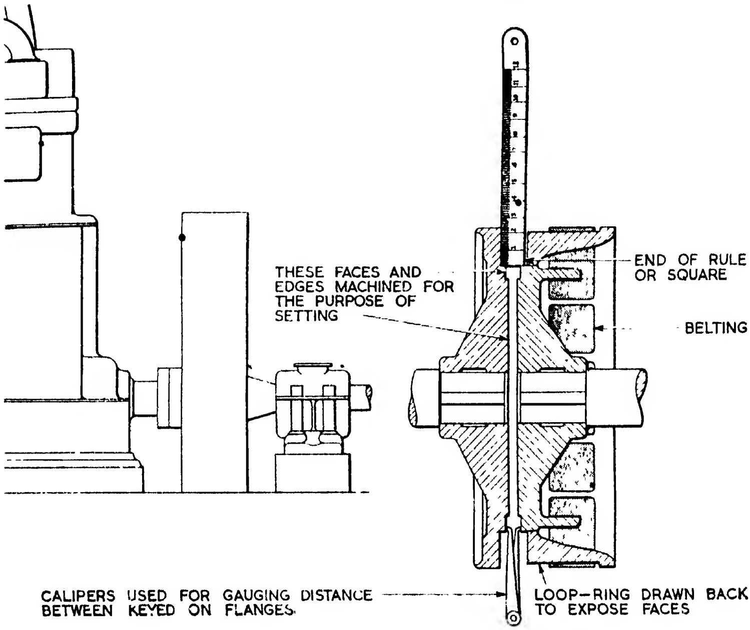


FIG. 4.—METHOD OF LINING UP FLEXIBLE COUPLING (*Ruston & Hornsby, Ltd.*)

Grouting Bolts into Foundation Block

Flat steel packing pieces of varying thickness and of a length and width to suit the particular size of engine should be provided by the makers and a set put next to each holding-down bolt. The steel levelling wedges are then removed, and the bolts are then grouted into the foundation block to within, say, 6 in. of the top. The grout should consist of equal parts of sharp sand and Portland cement, mixed with water so that it flows freely.

Erection of Tanks

Whilst the grout is hardening—and quick-setting cement will hasten this process—the erection of tanks and other accessories can proceed. There should be several days' work coupling up piping, etc., but the final position of the water-cooling tanks or fuel and oil tanks is best left until the pipes are completely finished. This will avoid any difficulties in mating flanges.

Next Step—Making Bedplate Level

As soon as the foundation bolts are fast, work on the engine must proceed, and other work must be left until the engine is properly grouted in. All the nuts

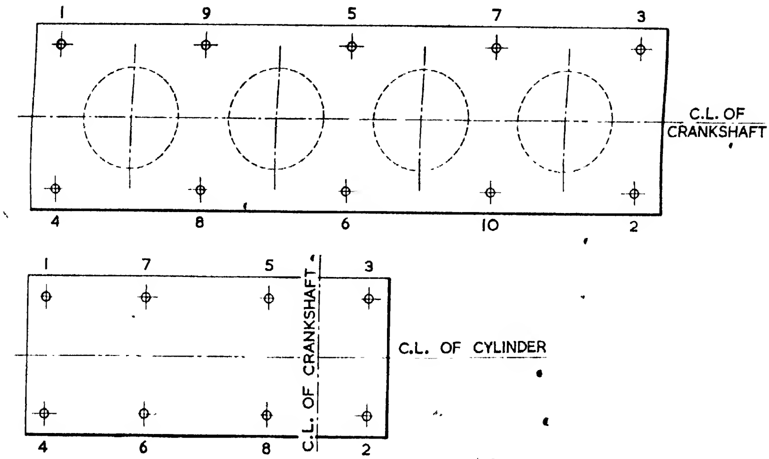


FIG. 5.—CORRECT ORDER FOR TIGHTENING HOLDING-DOWN BOLTS. (*Top*) FOUR-CYLINDER VERTICAL ENGINE. (*Bottom*) SINGLE-CYLINDER HORIZONTAL ENGINE

Bedplate nuts should all be tightened equally from corner to corner and across as shown This method avoids springing, and assists in lining up bearings correctly.

can now be tightened and a spirit level used in conjunction with a straightedge both longitudinally and across the bedplate.

Lining Up the Crankshaft

When the face of the bed has been made level by adjustment of the packing pieces at the various holding-down bolts, the crankshaft should be placed in position in the bearings, care being taken to see that all parts are thoroughly clean and that there is no dust or grit between the bearing shells and their housings. Particular attention should be given to the levelling-up process. If the makers are not erecting the complete installation or part of it, their services should at least be obtained if at all possible to check over the alignment of the crankshaft and bearings.

Method of Tightening Bedplate Nuts

The bedplate nuts should all be tightened equally, that is, from corner to corner and across, as shown in Fig. 5. This is important on a multi-cylinder engine bed in order to avoid springing and to assist in lining up bearings correctly.

Testing for Loose Crank-journal Bearings

It is as well to go over the crank journals with a spirit level and to try tapping the bottom halves of the bearings to find the loose ones which are not taking their share of the weight of the crankshaft. The bearing shells should

have been bedded into their respective housings at the works and each numbered, and care should therefore be taken to see that the bearings have not become misplaced from their correct positions.

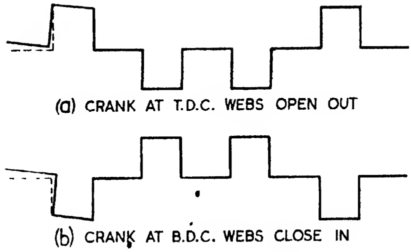


FIG. 6.—CRANKSHAFT WITH EXAGGERATED ALIGNMENT FAULT

Showing "breathing" of webs adjacent to a high bearing.

Crankshaft

With the top halves of the bearings tightened down evenly, the crankshaft, when lined up, should turn easily by hand.

If stiff, each top half should be inspected and scraped, if necessary, until free, and to give a running clearance as recommended by the makers, and determined by feeler gauges.

The bottom halves of the bearings should not be bedded to the crankshaft until the bedplate proper has been levelled. The makers will have previously run the engine on test and very little work should be required in this respect.

PLACING FLYWHEEL ON SHAFT

A Useful Tool for Erection and Maintenance

An easily made tool which is useful both in the erection and in the subsequent maintenance of the engine consists of a flat steel bar which can be secured to the crank web by two setscrews. This tool can also be used for removing and replacing the crankshaft bearings as described below. It is a good plan to have a fitment of this sort provided by the engine-makers.

Removing a Bottom Half-bearing

Should it be necessary to remove a bottom half-bearing at any time, the flat steel bar should be fitted, and the crankshaft turned slowly with the tool in position, thus bringing the half-bearing to the top, whence it is easily removable for attention. This saves lifting the crankshaft and obviates the possibility of damage to crankshaft or bearing. Care must, however, be taken to see that the width of the bar is less than that of the bearing section, so that the shaft may be turned without jamming.

The flywheel should next be lowered into the flywheel race of the foundation and supported on timber packings whilst the extension shaft is brought close up and the bolts fitted.

Fitting Wheel to Crankshaft Coupling Face

When the wheel is being fitted to the crankshaft coupling face, care should be taken to prevent straining of the shaft by overhanging the weight of the

wheel. The extension shaft should be supported by the outer bearing as soon as possible, so that the packing under the wheel can be removed. The same procedure applies if there is a dynamo shaft bolting up to the flywheel face, and in each case there should be shims, or liners, fitted to a total thickness of $\frac{3}{32}$–$\frac{1}{8}$ in. between the outboard bearing and its soleplate, in case future adjustments should be required.

Fitting Keyed-on Flywheel

If the flywheel is in halves or otherwise keyed on the crankshaft, extending in one piece to the outer bearing, then the latter can be fitted in position and its foundation bolts grouted in as soon as the bedplate and bearings are lined up. Generally, the quickest way of doing this is to place the soleplate on the concrete pedestal, and to hang the bearing on the crankshaft. The soleplate is then packed up to the bearing, with packing pieces similar to those used under the engine bed, until the bearing is properly supporting the shaft; the shims will, of course, have been fitted between the bearing and the soleplate before placing in position.

The holding-down bolts can then be grouted in, in a similar manner to those for the main bedplate.

After allowing sufficient time to set, the nuts may be tightened down and the crankshaft alignment checked and corrected if necessary. The flywheel should then be fitted, and if this is in one piece, it will be necessary to remove the crankshaft from the bedplate. This usually has to be done with the horizontal type of engine.

Checking Alignment of Complete Crankshaft

Checking the alignment of the complete crankshaft is now necessary, and for this purpose a stick gauge is inserted between the crank webs, and on rotating the shaft it is noted whether the webs open or close. A telescopic gauge can easily be made if not supplied with the engine, and will, in any case, be extremely useful for periodically checking over the crankshaft alignment.

Fig. 6 shows a line drawing of a multi-cylinder engine crankshaft, with exaggerated alignment fault. It can be seen from this that the crank webs of the throw, adjacent to a high bearing, will close when the throw in question approaches the bottom position.

Rectifying Non-alignment

The cylinder nearest the flywheel should be the first to have attention, and when the crank is on top dead centre, the gauge should be placed between the webs in the plane of the centre line of the crankshaft and adjusted to fit. The shaft is then given half a turn and the gauge tried between the webs. If slack, then the outer bearing is low, and it must be packed up slightly. If the gauge will not enter, then the packing of the bearing should be adjusted until the gauge is an exact fit in top and bottom positions. The horizontal alignment of the outer bearing should be checked in two positions, with the crank midway

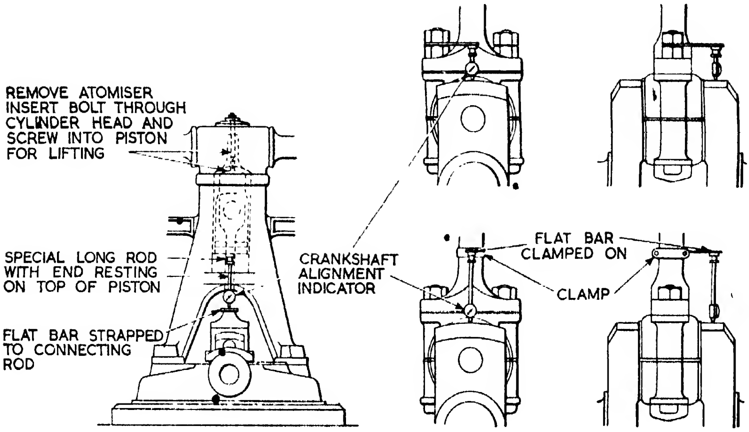


FIG. 7.—METHOD OF TESTING BEARING CLEARANCES WITH DIAL INDICATOR
(*Ruston & Hornsby, Ltd.*)

between top and bottom centres, and the bearing moved sideways until no breathing of the webs takes place.

On multi-cylinder engines the cranks should all be dealt with in turn and, when apparently correct, the whole shaft should be checked over to make certain that the adjustment to one throw has not interfered with that for the other cylinders.

Using Micrometer Dial Gauge

A spring-loaded gauge can be obtained which incorporates a micrometer clock. This gauge should be left between the webs and the plus or minus readings taken as the shaft is revolved. This type of instrument is sensitive and is easily used, and it is invaluable for showing at a glance the condition of the main bearings. The micrometer can also be adapted for measuring the clearances of the connecting-rod large- and small-end bearings.

The method of applying the dial gauge is shown in Fig. 7, which illustrates the method of testing clearances of both large- and small-end bearings.

Testing Shaft with Spirit Level

The crankshaft can also be tested for alignment with a sensitive spirit level, but it must be borne in mind that a slight allowance will have to be made for a deflection due to the weight of the flywheel. The level will read downwards towards the wheel both from the engine bearing and the outer bearing, the amount varying somewhat according to the distance of the centre of gravity of the wheel from the respective bearings.

Grouting in the Bed

The bed can now be grouted in, and the space left when grouting the bolts will serve to form a key. A wall of clay or wood of uniform height should be placed around the bed to retain the grout, and this should form the finished level of the grouting. The grout must be stirred well underneath with a flat rod, and in the case of large engines the main grouting should extend over two or three days to give time for settling.

The outer bearing is similarly grouted in, but not until the grout in the bolt holes has finished settling. The finished level of the grout should be above the bottom of the bedplate and soleplate, thus helping to make a good solid job.

Fitting Large-end Bearings to Crankshaft

The erection of accessories can now proceed once more. When the grout has sufficiently hardened to permit further work on the engine, the large-end bearings can be fitted to the crankshaft, after first making sure that all parts are absolutely clean and that no foreign matter is contained in the oil holes of the crankshaft, etc. The crank journals should be well oiled and the oil wiped round by hand before the bearing halves are bolted together. The compression plates should have been stamped for each cylinder at the makers' works, and these should now be checked over and placed in position.

Fitting Connecting Rod

The housing and gear can now be assembled, and, in the case of a horizontal engine, the piston and connecting rod. To fit the latter, the rod is fitted to the piston and then slung as near to the skirt of the piston as possible and the piston pushed into the cylinder liner by means of the large end of the connecting rod. It will be necessary to adjust the height of the piston very carefully to permit entry to the cylinder skirt.

Arrangement of Piston Rings

All the rings should be well oiled and the gaps arranged so that they do not come in line, for this would cause loss of compression. The liner should be well oiled and oil wiped round the piston before assembling. When closing the rings so that they enter the liner, care should be taken that the edges are not damaged, which would cause scoring of the piston or liner.

Take Care to prevent breaking Skirt of Piston

Before lifting the pistons of vertical or horizontal engines, a block of wood should be wedged between the under side of the connecting rod and the piston skirt. If this is not done, the piston may fall back with the gudgeon pin as fulcrum and thus break the skirt. In the case of vertical engines the packing is taken out as soon as the piston is in the vertical plane.

With horizontal engines the packing is kept in position until ready to connect up to the large-end bearing. A piece of wood should be placed under the rod and rested on each side of the engine bedplate to take the weight of

the rod. A length of pipe may be used for this, but there is a danger of it slipping forward towards the main bearings.

The back half of the large-end bearing is next mated up to the connecting rod and the bolts pushed through until the bearing half is held in position. The crankshaft is now turned so that the large-end journal is close up to the large-end bearing, the packing under the rod being adjusted so that the bearing half is in the path of the journal. The crankshaft can then be turned steadily until the journal touches the bearing, and the front half of this can be bolted up and tightened. This procedure is reversed for removing pistons or large-end bearings, and it is soon learned and quickly executed.

The same operation for a vertical engine is usually assisted by an eye bolt screwed into the piston crown, the cylinder head being, of course, removed, so that the piston and rod can be lowered into position from the top. For maintenance work it is generally possible to hold the piston in place by means of a bolt through the cylinder head whilst the large-end bearing is being removed.

To remove the piston and rod, the cylinder head is taken off and a bolt fastened in the piston, the large-end bearing is then disconnected and the piston and rod lifted out.

Care must be taken to see that the crankshaft is not turned whilst any bolts are fastened to the pistons, and it will also be necessary to see that the fronts of all rods are pointing to the front of the engine, for when the large-end connecting-rod bearings are bedded to the crankshaft journals they have to be lined up to the pistons, and to the small-end bearings, and this alignment may be lost if the bearings are reversed.

Erection of Valve Gear

The erection of the valve gear and camshaft, etc., can now proceed, and all brackets should have locating pegs, or else fitted studs or bolts to ensure correct alignment. Gearwheels which mesh will have the teeth stamped so that they can be meshed in the correct positions, and no mistake must be made here, otherwise the cam settings will be upset accordingly.

It cannot be emphasised too much that absolute cleanliness is essential in all stages of erecting and fitting major and minor parts. If the engine-house is not quite completed, all windows, doors, and openings likely to allow the ingress of dust and dirt should be blocked up with tarpaulins and sheets.

PIPEWORK AND TANK INSTALLATION

Pipework can now be permanently coupled up, screwed oil and water joints being made tight with hemp and a good jointing compound.

CLEANING.—Pipes, other than oil pipes, should all be blown out with compressed air, if available, and during the erection of the engine they should have their ends protected from dirt and damage. Oil pipes should be cleaned out with paraffin before fitting.

BENDING.—Pipes generally should be bent correctly before connecting up, in order to avoid strains on joints and flanges, when the engine is running.

All pipes liable to expansion must be free to move.

PACKING EXHAUST PIPES.—The exhaust pipes should be from 12 in. to 18 in. from any woodwork where they pass through the wall. It is advisable to pack the exhaust pipe in the wall with asbestos. To make a good job of this, bricks should be cut away to make a hole about 3 in. larger than the outside diameter of the pipe. Asbestos rope is then wrapped round the pipe to a thickness of about 1 in. and the remaining space grouted up with cement. This will permit expansion of the pipe, but the weight of the exhaust pipe and that of the exhaust manifold should be taken by means of roller supports.

JOINING VERTICAL PIPES.—Very special precautions are necessary with the joints of vertical pipes, since these are liable to transmit vibration. Makers' drawings and instructions should be rigidly adhered to in such cases; for instance, in the case of a vertical pipe connecting on to the flat bottom of an overhead water tank, or any pipe requiring a water joint where it passes through a roof.

Wherever possible, pipes should be carried outside the engine-room in a horizontal plane and should be amply supported.

DRAIN TAPS.—Drain taps should be fitted at the lowest points of all water pipes and provision made for emptying the pipe trenches of oil or water, as previously mentioned, whilst dealing with foundations.

LAGGING OIL LINE.—If there should be a length of fuel-oil line from the storage tank above the ground and exposed to frost, it is a good plan to lag this with straw or other suitable material.

Except when the fuel storage is overhead, it is advisable to install a semi-rotary or other suitable type of pump to assist the flow of viscous oil and to remove air locks from the system.

Fuel-oil Tank

A usual position for the fuel-oil tank is immediately outside the engine-room, supported by brick or concrete piers, the bottom of the tank being about 1 ft. 6 in. to 2 ft. 6 in. above ground-level.

OIL-LEVEL INDICATOR.—An oil-level indicator should be fitted inside the engine-room, and whilst this can be arranged for almost any position of the tank, it is as well to keep the arrangement of pulleys and gear as simple as possible.

UNDERGROUND TANKS.—To economise in space above ground, the tank or tanks can be buried underground, or else arranged in a pit with a suitably designed cover, fitted with a manhole so that a filling pipe from the wagon to the tank can be easily connected. Drainage of the pit should be provided for, likewise accessibility to pipe connections. The tanks should be well painted, and if to be buried and not enclosed in a pit they should be painted with special bitumastic paint. A fuel pump will be necessary in either of these cases.

FUEL TANK IN ROOF.—The fuel tank can be arranged in the roof of the

building, if this is suitably designed and sufficient space is available, but this is, as a rule, found more expensive than an outside tank. If, however, an overhead storage tank is decided on, then care should be taken that it is not so high that the fuel suppliers' wagons cannot pump to it.

Water-cooling Tanks—Test for Leakage before Installation

When several water tanks are supplied, as for the thermo-syphon system of cooling, each tank should be tested for possible leakage at the seams owing to damage received during transit. If each tank is tested separately, this saves loss of time and the trouble involved in disconnecting gear to rectify faults afterwards.

Piping Water from Pond or Sump to Jackets

With some systems of water cooling a pump draws water from a pond or sump and delivers it to the jackets. In these cases the suction pipe should have a gradual rise to the pump, otherwise airlocks may result if the pipe slopes towards the pump in any part of its length.

Piping to Cooler

If pumping hot water for delivery into a cooler, the length of suction pipe and the number of bends in it should be kept to a minimum, so that the total suction head, including friction, does not amount to more than about 10 ft. for a water temperature of, say, 150° F. Ten feet is not the maximum suction head allowable, but if exceeded it is possible that trouble may be experienced due to air bubbles, etc.; furthermore, the pump efficiency will be greater if the the suction head is kept low.

A foot valve and strainer of approved type should be inserted at the end of the suction pipe.

Installing Fuel-service Tank

The fuel-service tank in the engine-room should not be lower than the dimension given on the maker's drawing, whilst a little extra height will not do any harm. Cocks and valves in the various pipes should be easily accessible.

Water Drain from Compressed-air Receiver

There should be a water drain in the compressed-air receiver, which is usually included in oil-engine installations for starting purposes. If not possible or convenient for this drain to be taken from the bottom of the vessel, then a simple arrangement is sometimes obtained by fitting the drain valve or cock at the top of the receiver and leading an internal pipe from this almost to the bottom of the vessel. In this way, when the valve is opened, any accumulation of water is forced out by the air pressure, and when such a valve is used a bent pipe should also be fitted to deflect the stream of water and air downwards, thus avoiding the possibility of this being sprayed about the engine-room.

Position of Air Receiver

The air receiver should be placed as near to the compressor as possible, at the same time in a convenient position for the attendant and close to the engine, with a valve easily accessible for starting-up purposes.

Fuel-oil pipes should, where possible, be run in the same trenches as the exhaust pipes and close to them, thus assisting the flow of oil.

STARTING THE ENGINE

When all erection of pipework, etc., is complete, the various tanks and pipelines may be filled.

Care should be taken to see that fuel-oil and lubricating-oil tanks are thoroughly clean and dry before any oil is put into them.

Before opening up on any of the lines connected direct to the engine, an inspection should be made to see that all connections have been made correctly and that everything is clear away from the engine and that there are no packings, etc., left behind. These elementary precautions may seem futile, but they are well worth while for the sake of the short time that they take.

LUBRICATION.—A final precaution is necessary before attempting to start the engine, and this is to see that all lubrication services are thoroughly primed and that they are clear and the oil is actually reaching the place intended. It is not merely sufficient to see that the sight-feed lubricators are working; the point to verify is that the oil is going where it is wanted.

FOUNDATION.—Provided that the foundation is sufficiently well set, and in the majority of cases this will be so, the engine is now ready for starting. It should be borne in mind that the foundation should be about four weeks old if made with Portland cement, although this will depend a little on the quality of cement used and the size of the foundation. Obviously, if rapid-setting cement is used, this will reduce the time required.

READY TO START.—The engine may now be started, as directed by the makers in their instructions. These should be very carefully followed and the engine should get away without any trouble.

THE FIRST RUN.—This first test should be made without any load on the engine, and it should not last for more than a few minutes at the most. At the end of this time the engine should be stopped and all bearings felt for signs of undue warmth, and an inspection made to see that they are getting adequate lubrication.

THE SECOND RUN.—Next run for, say, ten minutes, and again inspect.

Running the Engine Under Load

If satisfactory, the engine may then be run up on load, commencing with a short run of about a quarter of an hour on quarter-load and subsequently increasing this to half-load if satisfactory. After running for about half an hour on half-load, if the engine seems to be going well and no symptoms of trouble have appeared, such as hot bearings, the load may be increased to full load and a run of, say, two hours' duration carried out. Several hours

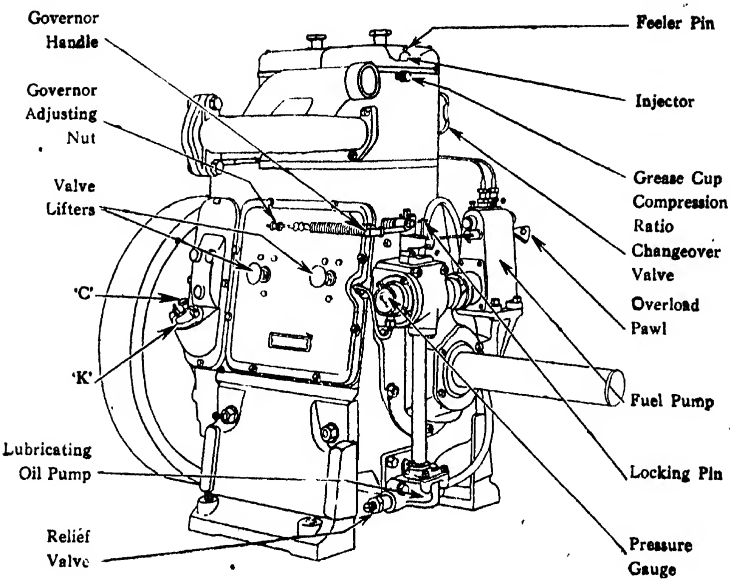


FIG. 8.—A TYPICAL SOLID-INJECTION DIESEL ENGINE
Showing various controls required for starting up. (*R. A. Lister & Co., Ltd.*)

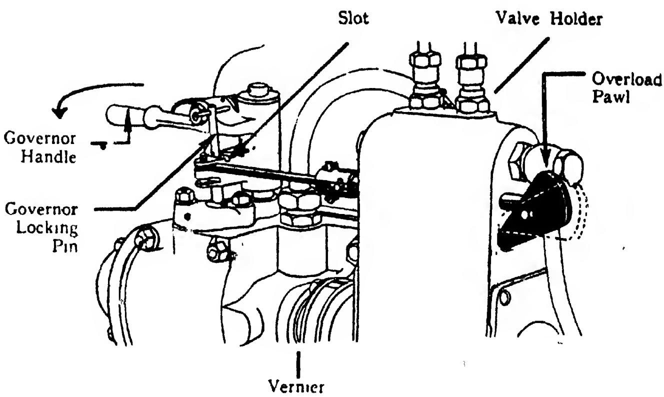


FIG. 9.—FUEL PUMP, OVERLOAD PAWL, AND GOVERNOR HANDLE (*R. A. Lister & Co., Ltd.*)

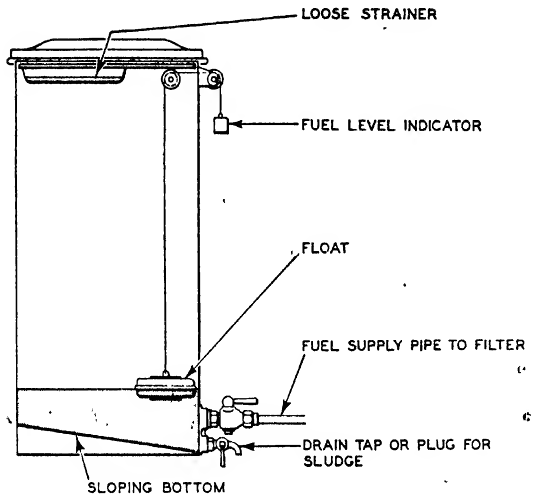


FIG. 10.—FUEL TANK

Showing drain tap and sloping bottom, with level indicator and float. (*R. A. Lister & Co., Ltd.*)

should be completed on load before any fuel-consumption tests or endurance tests are attempted.

With a multi-cylindered engine, tuning up and the distribution of load between the cylinders can be achieved with the aid of an indicator, but the same results can usually be obtained by a trained man using his physical senses alone.

DISTRIBUTION OF LOAD BETWEEN CYLINDERS.—If the exhaust branch from one cylinder is cooler than the others, this cylinder is not taking its fair share of the load. This also applies to the cooling-water outlets.

SMOKE FROM CYLINDERS.—The smoke from the cylinders is also a valuable indication of what is going on inside the engine, and the test cocks or plugs, usually provided on the exhaust manifold, enable the exhaust conditions to be observed. The loudest exhaust usually indicates the heaviest load, and is most often accompanied by heavier smoke. Dark-coloured smoke usually indicates overloading, whereas light-coloured smoke indicates underloading.

OVER-LUBRICATION.—A pale-blue smoke may indicate underloading, but it is also an index of lubricating oil in the cylinder over and above that normally necessary. It may be that the piston is being over-lubricated, or else that the scraper ring is not doing its work properly.

SMELL OF EXHAUST SMOKE.—A different smell attends the various loads.

BLOWING PAST THE PISTON AND ITS CAUSES.—Light loading may be coincident with crankcase pressure, when a blowing past the piston will be noticed. This may be due to badly fitting piston rings or rings stuck in their grooves, neither of which is likely to be the case in a new engine. If a local blowing is noticed, this may be caused by the rings working round so that the gaps are all in line. On some types and sizes of engines where this trouble has occurred, the makers peg the rings with the gaps staggered, thus permanently overcoming this trouble. If the pistons have been put in as instructed it is unlikely that the

gaps are in line, and it is more likely that there is a slight distortion, which may disappear when the working parts bed down together.

How to get an Accurately Tuned Engine

A clear exhaust should be the aim of the engine attendant. To assist in this the fuel injectors should be made interchangeable and the springs and needle valves should be identical, whilst on some makes the lift, or spring pressure, can be adjusted to correspond with that of other cylinders.

Fuel-oil pipes from fuel pumps or distributors should all be of the same length, and for a neat arrangement of these pipes coils can be made of the surplus length of those nearest the pump.

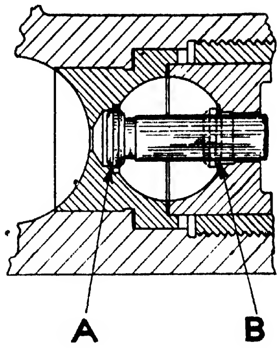


FIG. 11.—CHANGE-OVER VALVE

Valve clearances must be uniform and to the makers' recommendation.

Air filters must be clean to prevent obstruction to the air charge upon the opening of the inlet valves.

Checking Fuel Consumption

Finally, the fuel consumption should be checked, and this is the ultimate proof of an accurately tuned engine. The fuel consumption guaranteed by the makers should at least be obtained, for these figures are mostly covered by a margin.

Causes of Heavy Fuel Consumption

If the consumption is heavy, then the cam settings should be checked over, the governor inspected for steadiness, and the atomisers or fuel injectors tested for a correct spray without "dribble."

Dirty Exhaust

A dirty exhaust may be caused by a fuel injector not working properly, late timing of the fuel injection, or by a choked air filter.

The needle valves of the injectors or atomisers should be an easy running fit in their sleeves or in the body, and should seat correctly.

STARTING UP A TYPICAL SOLID-INJECTION DIESEL ENGINE

The following notes on starting up and stopping a solid-injection Diesel engine apply to a typical hand-controlled engine of this type. The engine, as illustrated in Fig. 8, is fitted with a compression change-over valve, which ensures a very high compression for starting, thus giving greater heat of compression for the ignition of the fuel.

Preparing to Start

Before actually starting the engine, the following procedure should be adopted:

(1) Open the oil filler *K* (Fig. 8) and pour in lubricating oil of the correct quality, until the level in the sump reaches the mark on the dipstick *C*. Close the oil filler after filling.

(2) Fill the water-cooling system with water, avoiding " hard " water whenever possible.

In the case of tank cooling, open the three-way tap in the bottom water-tank connection so that water can flow freely from the tank into the cylinder jacket. The water should be kept above the top connection in the tank.

(3) Now fill the fuel tank, using a fine gauze strainer.

The fuel tank should be fitted so that the bottom of the tank is approximately $1\frac{1}{2}$ ft. above the fuel pump. To collect the sludge, etc., deposited by the fuel, a good method is for the fuel tank to be fitted with a sloping bottom leading to a drain tap, which should be used to flush out the sludge from time to time, as found necessary. To prevent sludge from entering the fuel pipe, the fuel outlet connection should be placed at least 2 in. above the bottom of the fuel tank and the drain tap at the lowest point.

(4) After filling the fuel tank, turn on the fuel tap on the tank.

Priming the Fuel System

(5) Next prime the fuel system. To do this, prime the filter by unscrewing the vent screw on top of the filter with a spanner until the oil flows freely through. Then retighten the screw.

As it is highly important that all air should be removed from the fuel pump and fuel valve, in order to prime the fuel system, disconnect the fuel-injection pipe from the delivery valveholder on the fuel pump by unscrewing the union. It is necessary to see that the governor lever handle is in the " Stop " position.

Then remove the delivery valveholder and spring, and with the fingers slightly raise the delivery valve from its seating; as soon as this is done fuel should appear. The delivery valve should be held off its seat until all air bubbles are out of the system and until a solid column of fuel oil appears. Then replace the delivery valveholder and spring and tighten down the holder carefully and not too vigorously, so that the body of the fuel pump is not distorted.

The fuel-injection pipe should now be connected again to the fuel pump, but in order to see that no air is present in the fuel pipe loosen the fuel-pipe union at the injector. Then place the governor lever handle in the " Start " position by withdrawing the release pin. Put the starting handle on the crankshaft and push in the compression release knob on the crankcase door, in order to lift the exhaust valve. Then turn the crankshaft until fuel free from air bubbles appears at the fuel-pipe injector union. Tighten the fuel-pipe union.

Finally, in order to make absolutely certain that a solid column of fuel is reaching the fuel valve, and that no air is present, place a finger on the feeling pin in the top of the injector, and if fuel is present without air, a definite creak

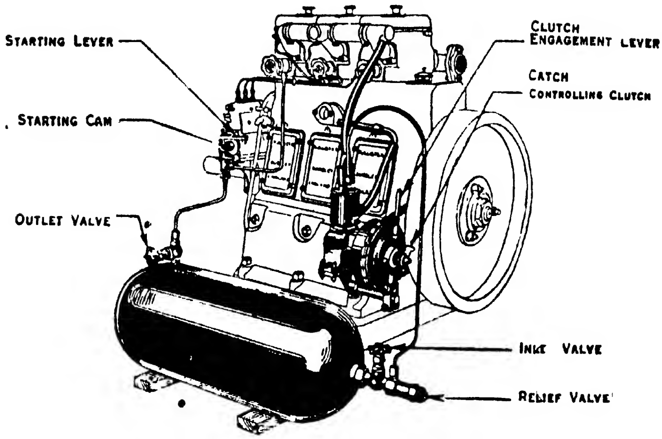


FIG. 12.—ENGINE FITTED FOR COMPRESSED-AIR STARTING—SEPARATE AIR-COMPRESSOR TYPE (*R. A. Lister & Co., Ltd.*)

will be heard in the fuel valve, and the lift of the valve will be felt by the finger.

On multi-cylinder engines this procedure must be carried out with each pump element.

(6) After priming the fuel system, see that the compression release knobs are pushed in.

(7) Set the governor handle to START.

(8) Lift the overload pawl to allow governor lever to move to its maximum position. This pawl will return to normal as soon as the engine starts.

(9) Screw the compression ratio change-over valves inwards tight on to their seats.

To Start

(1) Grip the starting handle firmly, ensuring that it is fully engaged with the crankshaft end, and turn smartly. When a good speed has been attained on the flywheel, pull out the valve lifters, when the engine should immediately fire. On multi-cylinder engines the other valve lifters should be pulled out immediately after.

(2) As soon as the engine has attained its normal speed, open the compression ratio change-over valve by screwing the handwheels outwards until they come to a stop.

(3) When the change-over valve is screwed in, it beds against the seating at *A*, Fig. 11, and when screwed out against seating at *B*. These seatings should be kept free from carbon. This can be done by turning the valve handwheel backwards and forwards once or twice before leaving in the final position, tight up.

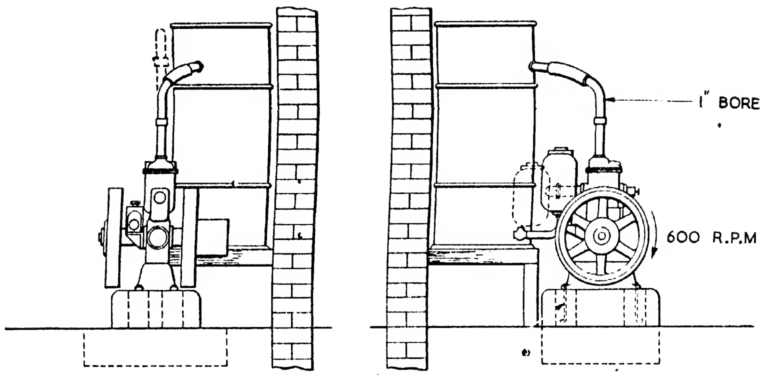


FIG. 13.—TYPICAL INSTALLATION DRAWING OF A 5-B.H.P. PETROL ENGINE—SIDE AND END ELEVATION

Speed Regulation

Should a slight adjustment in speed be required, this can be made by screwing the adjusting screw inwards to increase the tension on the governor spring. This increases the speed, and by screwing out the adjusting screw the tension on the spring is eased and so the speed is dropped slightly.

To Stop Engine

Turn the governor locking pin until it falls into the slot. Pull back the governor handle in the direction of the curved arrow (Fig. 9) and the engine will stop.

Do not turn off fuel-supply tap except in case of emergency. The turning "off" of this tap may lead to air locks in the fuel-supply system and the necessity for repriming it. Never try to stop the engine by lifting the exhaust valve.

Compressed-air Starting

To start the engine on a plant fitted with an air receiver and air compressor for compressed-air starting (Fig. 12), carry out the instructions already given with regard to filling up with fuel, oil, and water, etc., and set the governor lever handle in the starting position. Screw in the handwheels of the compression change-over valves, except that fitted with the starting air valve, and put the engine on top dead centre of No. 1 cylinder, as marked on the flywheel. In this position the ball pad on the starting lever should be on the top of the starting cam. Put the engine on full compression by pulling out the exhaust-valve lifters. Open the outlet valve and press the starting lever, when the engine should start immediately. Screw out the handwheels, close the outlet valve, and proceed to charge the air receiver.

INSTALLING A PETROL-ENGINE LIGHTING SET

Stationary engines are usually mounted on a concrete base, and this should be from 6 in. to 10 in. above floor-level, according to the size. It is the usual practice of manufacturers to provide a drawing with the engines they sell, and this gives all the information necessary for the construction of the foundation.

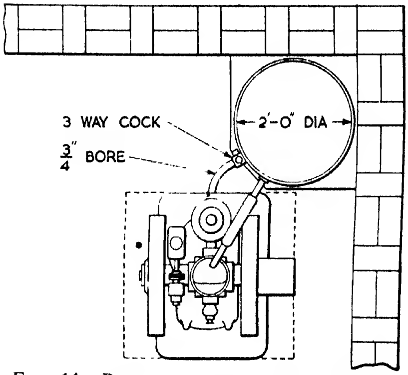


FIG. 14.—PLAN OF TYPICAL PETROL-ENGINE INSTALLATION

A Typical Installation Drawing

An example of a typical installation drawing is shown in Fig. 13. This shows a small single-cylinder petrol engine with its foundation, cooling tank, and exhaust silencer set out to comply with the manufacturer's recommendations.

Concrete for Foundations

The best composition of the concrete for the foundation block is one part of Portland cement with four or five parts of washed ballast.

After the foundation is constructed, it should be allowed to stand for a few days before any attempt is made to start erecting the engine. In very hot or dry climates it is desirable to keep the foundation flooded with water during this period. The holes for the foundation bolts should be much larger than the diameter of the bolts to facilitate erection.

The alignment of multi-cylinder engines on the foundation is a matter of extreme importance.

Lowering the Engine into Position

The engine should first be carefully lowered to within about 1 in. of the concrete and allowed to rest on strips of steel about 8 in. long, 2 in. wide, and of varying thickness. These strips or packing pieces should be near the foundation bolt holes, and the engine should be levelled by the insertion or removal of strips until it is perfectly level, when the foundation bolts can be grouted. After an interval of a day or so, the concrete in the bolt holes will have set, and the nuts on the bolts can be pulled down tightly.

Levelling Bedplate

The level of the bedplate should then be carefully tested with a spirit-level. If any deflection is observed, the inequality must be corrected by the removal of some of the packing strips until the bedplate is level at all points.

When this stage is reached, the foundation can be finally grouted up with a composition of two parts of fine sand and one part of cement.

Arrangement of Belt Drive

If the engine is to drive by belt on to fast and loose pulleys, these should be arranged in such a way that when driving on to the fixed pulley the belt is near the engine bearing. This applies particularly when the engine pulley is overhung, e.g. when there is no outboard bearing.

The underside of the belt should be the driving side whenever possible, as the arc of contact is thereby increased by the sagging of the slack side. The pulleys should be arranged to give a belt speed of about 3,500 ft. per minute and care should be taken that the belt used is capable of transmitting the necessary power. The life of a belt will be prolonged and its driving powers increased by keeping it in good working order.

Direct Coupling

It frequently happens that engines are to be directly coupled to some other machine such as a dynamo, pump or air compressor, and that such machines are not mounted with the engine on a combined baseplate. In such cases, the procedure outlined above for engine levelling should be applied to the machine which is to be driven.

Aligning Direct-coupled Shafts

Should the driven machine be rigidly coupled to the engine, it is of the utmost importance that the shafts of both should be in the same horizontal plane in order to prevent malalignment. If they are not, there is a danger of the crankshaft fracturing due to fatigue set up by the alternating stresses generated.

To obtain correct alignment, the two halves of the coupling should be brought together and a feeler gauge inserted between the coupling faces. If the same reading is obtained at intervals of 90° round the coupling, the bolts can be tightened up.

If a flexible coupling is used, the results of incorrect alignment will not be dangerous, but even so the stresses will have to be absorbed at some point and the same care should be taken to ensure that the two shafts are co-axial.

Alignment Indicator

When the two machines are finally connected, the alignment should be checked by means of an alignment indicator. This simple instrument can be bought for a fraction of the cost necessary to defray the cost of a new crankshaft, particularly if the engine be of medium or high power. The check is made by inserting the indicator between the webs of the crankshaft and noting the readings at intervals of 90°.

Malalignment will be most noticeable between the webs nearest the flywheel, and a difference of 0·002 in. should be investigated and corrected.

It is a commendable practice periodically to check the alignment because the foundations may subsequently settle, and disturb the position of the engine

and driven machine sufficiently to move the two shafts out of alignment. The importance of correct positioning will be appreciated when it is remembered that if the engine speed is 800 r.p.m. there are 384,000 alternating stresses in a running period of eight hours.

Cooling Equipment

Most stationary engines are cooled by the tank thermo-syphon system, and this is an excellent system in all cases where there is no objection to space which the water tank or tanks must occupy. If the water tanks are not supplied by the engine manufacturers, the user should be advised by them regarding the required capacity. For normal conditions a capacity of 8 or 10 gallons per b.h.p. of engine power is usually allowed. If the engine is working continuously for long periods, or is installed in a tropical climate, this capacity should be increased by 50–100 per cent.

Installing a Thermo-syphon System

In a thermo-syphon system, the water tank should preferably be installed out of doors, and the top of the tank should be well above the top of the engine. The size of water piping should be that recommended by the manufacturers and the pipe connecting the engine outlet to the tank should have a gradual rise. The water in the tank should always be above the pipe from the engine, otherwise the circulation will cease unless the natural thermo-syphonic action be assisted by a water pump. If there is no pump in the system, it is desirable to connect the main supply to the tank, and fit a float valve to maintain the level.

"Run Through" Cooling System

If there is an ample supply of water available, water tanks need not be used, as the water can be allowed to run through the engine to waste. This is known as the "run through" system, and when used the water should be available at a pressure of not less than 10 lb. per square inch, which is equal to a head of about 20 ft. With this system, from 1½ to 3 gallons of water per b.h.p. hour are required, according to the temperature of the water being used.

"Hard" Water

In some localities the water is "hard," that is, it contains much mineral deposit held in suspension. Such water will deposit lime if its temperature when leaving the engine exceeds about 150° F. This precipitation in the water spaces rapidly reduces their area and consequently the engine "overheats." The remedy is to remove the scale from the water spaces and increase the velocity of the water to reduce the temperature rise.

Radiator Cooling

Some engines are cooled by a radiator similar to the motor-car, and usually the circulation is assisted by a water pump. If there is no pump, the water in the radiator should be at a higher level than the top water pipe.

Exhaust and Exhaust Gases

The disposal of the exhaust gases is a matter about which some care must be taken. The piping should be as short as possible, and the atmosphere end of the pipe should be fitted with a bend or cowl.

In thickly populated areas, the exhaust should be well silenced, and in such cases the use of an auxiliary silencer or pit is recommended. If the bore of the exhaust piping is too small, back pressure will be set up and this is detrimental to the engine. The bore of the piping should not be less than that recommended by the engine builders.

Fuel Tank

Most stationary engines have a gravity-feed fuel supply and the fuel tank should be slightly above the level of the carburettor in order that there may be a good head of fuel at the delivery point. The fuel tank should be fixed in an accessible position and where there is no likelihood of water reaching the fuel.

The Initial Start

TIGHTEN ALL NUTS AND JOINTS.—After the engine has been erected and the piping connected, all external nuts and piping joints should be tightened.

FILL WATER AND FUEL TANKS AND LUBRICATOR.—The water and fuel tanks should be filled, and the lubricator should be filled with the correct grade of lubricating oil. The engineer should go round the engine with an oilcan to fill up the sundry oil holes; and if the main bearings are ring-oiled, the oilwells should be filled up to the correct level.

TEST FOR OBSTRUCTIONS.—The engine should be given a few turns by hand, with the compression cocks open, to ensure that there are no obstructions. It is wise to make sure that the water is reaching the cylinders, and that there are no obstructions in the piping. The lubricating pipes should be primed, and care should be taken that these pipes also are free from obstructions.

LUBRICATION.—If the lubrication is of the drip-feed type, set the drops per minute to the figure recommended by the manufacturers.

TURN ON FUEL.—Turn on the fuel and lightly flood the carburettor.

SWITCH ON AND START.—Switch on the ignition, and if the engine is of the hand-starting type, it should start after a couple of turns.

TIGHTEN JOINTS.—When the engine is "away," let it run light for a time, and after it has warmed up, go round the engine again with spanners and pull down all bolts and joints tightly. It is frequently found that joints exposed to heat will need tightening when the engine is thoroughly warm.

"TUNE" OF ENGINE.—A mental note of the "tune" of the engine should be made. The sounds made when running are a reliable guide to the engine's performance.

FAULTY ENGINE.—So far it has been assumed that the engine has been started up quickly and easily, and this should be the case with a new engine, although there are exceptions. If a start is not effected after a few smart turns, a check of the engine should be carried out in order to try to locate the reason.

OPERATION AND MAINTENANCE OF DIESEL LOCOMOTIVES

DURING the last twenty years there is no doubt that the diesel locomotive has come into its own, and is being installed in increasing numbers with every year that passes.

Diesel locomotives fall naturally into three classes:

(1) Narrow-gauge machines weighing up to 14 tons used for surface working in quarries, brickworks, plantations, on public works' contracts, and similar applications. There is also a large field where this type of machine is fitted with exhaust conditioning apparatus for work underground in metalliferous and coal mines and the like.

(2) Shunting locomotives working on the normal main-line gauge of the country in question. These are used largely on the internal railway systems of factories, steel works, docks, harbours, electricity stations, gas works, and on similar duties.

Generally speaking, the smallest locomotive of this type weighs $7\frac{1}{2}$ tons and the largest 45 tons approximately.

(3) Locomotives owned and operated by main-line railway authorities, and these can roughly be classed as follows:

(*a*) Fast main-line locomotives largely used on express passenger work.

(*b*) Smaller mixed-traffic locomotives.

(*c*) Shunting locomotives stationed permanently in marshalling yards.

A further division can be made with reference to the manner in which the power is transmitted from the engine to the rail wheels. In common use to-day, two types of transmission are found—one mechanical and the other electric. Experience shows that the majority of the locomotives coming within categories (1) and (2) are fitted with mechanical transmission, and the following notes relate to an engine of this type.

Another subdivision discloses a difference between the types of mechanical drive, particularly as regards the gearboxes, which are operated either manually or hydraulically.

CONSTRUCTIONAL DETAILS

It may be an advantage now to consider the design of a specific locomotive, for which purpose we are taking a medium-weight machine manufactured by Messrs. Ruston & Hornsby, Ltd., of Lincoln.

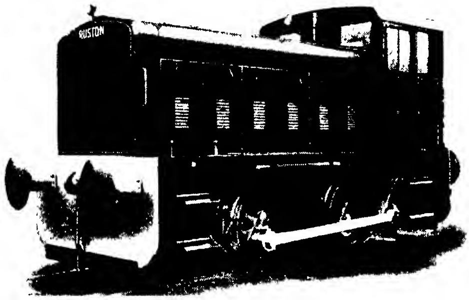


FIG. 1.—MEDIUM-WEIGHT (28-TON) LOCOMOTIVE
(*Ruston & Hornsby, Ltd.*)

The locomotive is shown in Fig. 1, and can be subdivided into the following major components:

(1) Power unit.
(2) Gearbox.
(3) Frame and running gear.
(4) Engine bonnet, cab, etc.
(5) Compressed-air system comprising compressors, receivers, and piping.

Power Unit

This, as the illustration shows (Fig. 2), is a vertical-type six-cylinder, four-cycle diesel engine, running at a maximum speed of 1,250 r.p.m., with a speed range down to 500–550 r.p.m.

The housing is fitted with renewable wet-type liners. The cylinder heads carry the inlet and exhaust valves with their attendant rocking levers actuated through hardened-steel tappets operated by a camshaft carried in the housing and driven by chain from the main crankshaft.

Each cylinder head also carries a fuel injector which receives a supply of fuel from the fuel pump of the monobloc type carried on a bracket on the engine housing. The purpose of the injector (illustrated in Fig. 3) is to convert the solid fuel to a fine mist or spray, which ignites, due to the temperature rise in the engine cylinder during the compression stroke.

The crankshaft runs in main bearings lined with anti-friction metal carried in steel shells, whilst the connecting-rod bearings are of similar construction.

The connecting rod small-end bush is of phosphor-bronze, working on a floating gudgeon pin in the aluminium-alloy piston.

Starting is effected by compressed air delivered in correct sequence to each cylinder through the means of starter valves controlled by a distributor operated from the camshaft.

Cooling water is circulated through the engine system and radiator by means of a centrifugal pump driven from the main crankshaft, the pump forming an integral part of the power unit. A thermostat is also fitted to give appropriate temperature control.

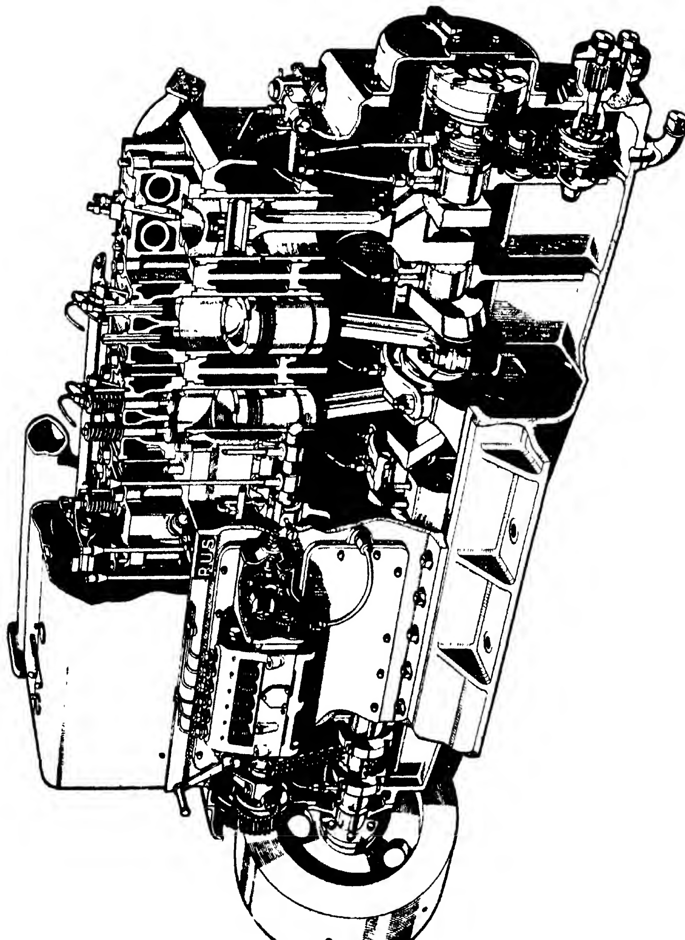


FIG. 2.—VERTICAL-TYPE SIX-CYLINDER, FOUR-CYCLE DIESEL ENGINE

Cylinder heads, housings, and bedplate are all separate castings, the bedplate carrying the crankshaft. (*Ruston & Hornsby, Ltd.*)

The flywheel is located at the driving end of the engine, and carries a flexibly mounted shaft which takes the drive to the coupling on the gearbox.

FIG. 3.—RUSTON MARK 37 FUEL INJECTOR (*Ruston & Hornsby, Ltd.*)

Gearbox

The gearbox fitted to this type of machine is the constant-mesh totally enclosed oil-operated type arranged to give four speeds in either direction of travel, and is made by Ruston & Hornsby, Ltd., under S.L.M. patents. The sectional view of Fig. 4 shows clearly the method of construction.

A primary shaft (*S*) is coupled to the power unit as described, and carries four gears, all of which are in constant mesh (*D*), with four secondary gears carried on shaft (*T*). These secondary gears (*U*) are of special construction as follows:

The outer parts of the gear are made in two halves screwed together with teeth cut on their peripheries meshing with the four gears on the primary shaft. These steel outers run freely on the inner clutch members (*V*). Inside each secondary gear are two phosphor-bronze sliding clutch plates (*V*) telescoped together and machined on their outside faces, with annular ridges and grooves of tapering cross section registering with similar grooves on the inside of the steel gears. These clutch plates are splined and free to slide on shaft (*T*).

To engage a gear, oil under pressure is admitted through shaft (*T*) to the appropriate clutch plates (*V*), which are thus forced apart to engage the secondary gears (*U*). The secondary shaft (*T*) carries a bevel wheel (*W*), which transmits the drive to the reversing gears and thus to the final drive shaft which carries the jackshaft of the locomotive.

The power is taken to the jackshaft as follows:

Through the appropriate gear on the primary shaft to the steel outer part of the gear on the secondary shaft; this being solid with the internal phosphor-bronze part, the drive thus passes to the secondary shaft and so to the final drive.

To change gear, the control valve is arranged so that it can supply oil to any of the four appropriate secondary gears, being designed in such a way that until the oil pressure is released from one set of secondary gears it cannot be transferred to another set.

Oil pressure is supplied by a gear-type oil pump driven from the primary shaft of the gearbox, a relief valve being fitted to the oil-pressure system and operated by the change-gear lever in the locomotive cab. Its purpose is to reduce the oil pressure when engaging first gear, thus ensuring smooth take off or accurate control of the locomotive during inching, and it is also used to disconnect the drive by relieving the oil pressure on any gear clutch during the application of the airbrakes with which the machine is fitted.

The forward and reverse bevel gears are driven from the bevel on the

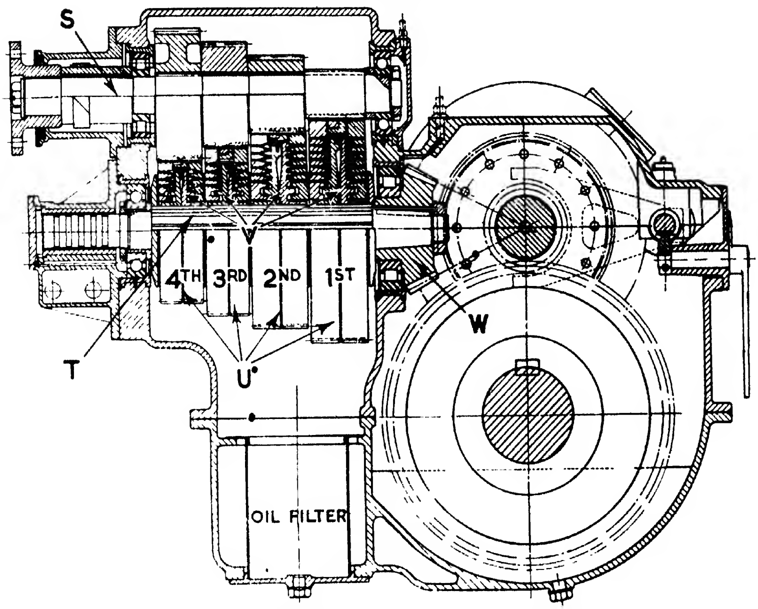


FIG. 4.—SECTIONAL VIEW OF GEARBOX FITTED TO THE RUSTON MEDIUM-WEIGHT LOCOMOTIVE
(*Ruston & Hornsby, Ltd.*)

secondary shaft, and each of these former are fitted with dog clutches pressure operated by means of a control valve situated in the cab.

The gear-change control and the forward reverse controls in the cab are mechanically interlocked, so that it is impossible to operate the forward and reverse control, while any one of the main-gear clutches is engaged. Furthermore, the forward and reverse control carries a neutral position in which it is impossible to engage the main clutches: thus, when not attended, the locomotive can be left in a perfectly safe condition.

Main Frame and Running Gear

The locomotive frame consists of heavy steel plates and cross members welded together to form a single unit able to withstand the heavy shocks and stresses encountered during normal shunting operations. Combined in this frame is the seating to carry the power unit, also carrying faces on the side members to take the gearbox.

The axle boxes are carried in horn guides fitted to the main-frame side plates by means of fitted bolts, and arranged in such a way as to be easily removable when renewal is necessary.

The axles carrying the rail wheels are, of course, in the axle boxes, and are

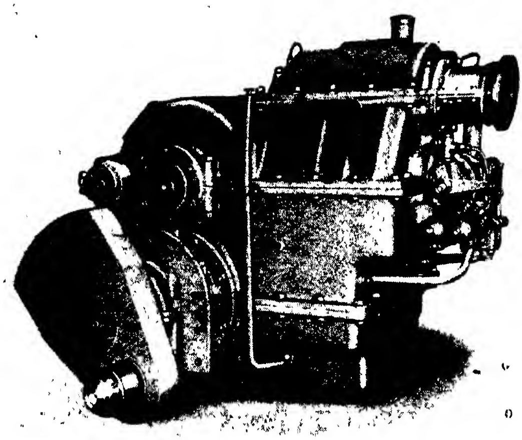


FIG. 5.—EXTERNAL VIEW OF GEARBOX FITTED TO THE RUSTON MEDIUM-WEIGHT LOCOMOTIVE

The casing itself is supported in the frame by two horseshoe plates, which form an integral part of the gearbox.
(*Ruston & Hornsby, Ltd.*)

suspended by means of laminated springs carried on steel hangers slung inside the frame.

The drive to the rail wheels is taken from a jack crank through the means of forged-steel connecting and coupling rods.

Compressed-air System

As the locomotive relies on the compressed air for starting the power unit, operating the reverse mechanism of the gearbox, applying the brakes, and

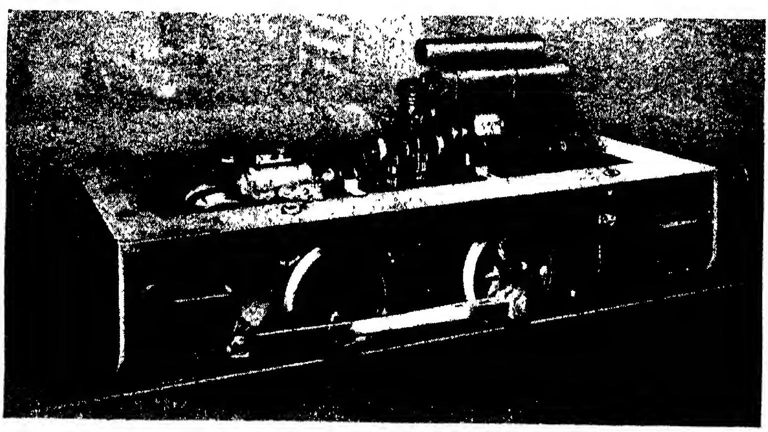


FIG. 6.—MEDIUM-WEIGHT LOCOMOTIVE FRAME

The illustration shows the locomotive frame carrying the power unit, gearbox, and also the running gear. (*Ruston & Hornsby, Ltd.*)

operating the sanding mechanism, provision must be made for an adequate and constant supply of air. This is achieved by mounting on the locomotive an air-cooled compressor driven by belt from the extension shaft between the engine and gearbox, and constantly running when the main engine is in operation. In addition, a separate compressor driven by an independent petrol engine is also mounted under the bonnet, the main function of which is initially to charge the engine-starting air receiver, but it is also useful during periods of maintenance, receiver inspection, and the like. Piping from this second compressor set is, of course, arranged in such a way so as to deliver air to the main air piping system normally used by the compressor driven from the main engine.

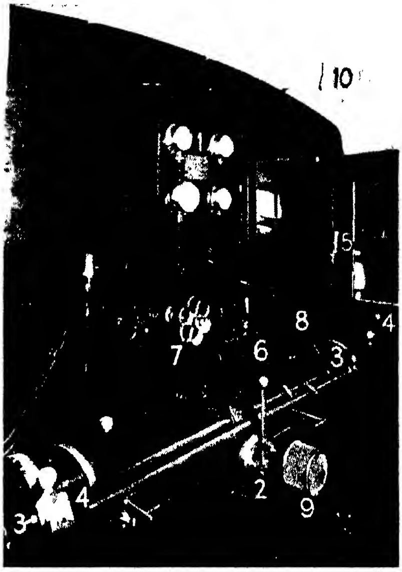


FIG. 7.—LOCOMOTIVE CAB INTERIOR

(1) Lubricating oil and air pressure gauges.
(2) Forward reverse control.
(3) Gear change lever.
(4) Engine-speed control.
(5) Air-brake control duplicated on either side of the cab. Other handle out of picture on the left.
(6) Sanding control.
(7) Engine air-starting valve.
(8) Electric switch-box.
(9) Cab heater
(10) Pneuphonic horn control.

(*Ruston & Hornsby, Ltd.*)

MAINTENANCE

For successful maintenance the possession of a suitably equipped locomotive shed is almost a necessity. This should be adequately lighted and equipped with an inspection pit, lifting gear, and work bench with vices. It is also desirable that the shed should be heated during the winter months, and provision should be made for the storage of dry sand for replenishment of the sand boxes in the locomotive.

An injector testing pump of the type illustrated in Fig. 8 is also extremely useful, whilst reasonable lock-up cupboard facilities for the storage of certain essential spares, such as injectors, joints, filter elements, and brake blocks, should be arranged.

Examination and Maintenance Periods

Examination periods for the various parts of the locomotive naturally depend upon the working conditions, but experience shows that the following is a reliable

approximate guide. Users find, however, through experience, the most suitable periods to meet their own conditions, and these should be carefully enforced.

Daily, 8–12 *hours*

Power Unit: Check engine lubricating-oil level.
Check fuel-oil level.
Top up lubricators and grease cups.

Locomotive: Top up lubricators and grease cups.
Examine water level in radiator.
Check compressor oil level.

Two to three days, 20–36 *hours*

Locomotive: Drain starting air receiver of water by means of cocks provided.
Drain brake air receiver of water by means of cocks provided.

Weekly, 50–60 *hours*

Power Unit: Clean lubricating-oil filter.
Clean fuel-oil filter.
Clean air filter.
Service air filter.

Locomotive: Check gearbox oil level.
Use oilcan to all joints, buffer stems, etc.

Monthly, 200–300 *hours*

Power Unit: Drain and refill engine lubricating-oil sump after cleaning by flushing out.
Remove injectors and test spray. If in order, replace without interference. See detailed instructions for cleaning injectors if this is necessary.
Inspect fuel system to see that all joints and connections are tight.
Drain and clean out fuel filters, washing the elements in clean petrol or paraffin.
When engine is running, make sure that starter valves are seating properly, and not leaking. Leakage is indicated by warm pipe.

Locomotive: Drain and refill compressor oil sump.
Examine all driving belts and adjust if necessary.
Examine all important nuts and bolts and joints for tightness.
Examine brake shoes, and adjust if necessary.

Every two months, 400–600 *hours*

Power Unit: Clean injector filters.
Remove engine covers and see that all bearing nuts are tight and split pins intact.
Check for excessive lift of bearings by means of bar under flywheel, or connecting-rod large-end bearings.
Examine electric battery used for lighting system.

Every three months, 600–900 *hours*

Locomotive: Drain gearbox oil sump and refill with new oil.

Every four months, 800–1,200 *hours*

Power Unit: Remove water-jacket cover and remove any mud or scale.

Every nine to twelve months, 1,800–3,600 *hours*

Power Unit: Remove cylinder heads and clean out water spaces.
Draw each piston and carefully clean out oil-return holes.
Examine and adjust all bearings.
Examine and grind in, if necessary, all inlet and exhaust valves.
Wash out all lubricating-oil pipes.
Check all nuts, including those in the timing gearcase.
Clean out exhaust manifold and piping.
Clean and grind air-starter valves.

Locomotive: Check all nuts.
Examine bearings.
Remove gearbox covers and examine gears, clutches, bearings, pump, etc. See that all locknuts and screws are tight before replacement of covers.
Remove air-brake cover and examine piston and ring.
Check axle boxes, horn guides, and running gear generally, including coupling and connecting-rod bushes.

CARE OF INJECTION EQUIPMENT

It is true to say that the satisfactory running of a diesel engine is largely dependent upon the condition of the fuel injectors. Maximum efficiency in operation can only be expected if the injector is delivering the fuel oil in such a

FIG. 8.—RUSTON INJECTOR TESTER

This pump is designed for use in cleaning, checking, and testing injectors after reconditioning.

(*Ruston & Hornsby Ltd.*)

'orm as to produce the most efficient combustion. When an injector is not running well, the cause is generally due to the presence of dirt or other foreign matter.

To assist users to keep this equipment in perfect condition, a testing pump ınd a set of reconditioning tools are invaluable. There are proprietary equipments of this nature obtainable, and some makers supply them themselves.

A few notes on the use of these accessories may be worthwhile.

Testing Pump

The construction of this can be plainly seen from the illustration (Fig. 8). ts uses are briefly expressed as follows:

1) To clean the injector by pumping clean, light oil through it.
2) To check periodically the quality of the atomising.
3) To check the quality of the spray after the injector has been dismantled either for reconditioning or for fitting new parts.

By using the pump for purposes (1) and (2), the dismantling of the injector s reduced to the very minimum.

Method of Use

Clean, well-strained fuel oil is poured into the container, and the pump is 'reely operated until its discharge is free of air. The adapter nut is then connected to the injector.

The necessity for using clean fuel oil, free from any foreign matter, is particularly emphasised, as the greater part of injector wear and troubles are lue to the use of unclean fuel oil.

At first, fuel oil should be forced through the injector for the purpose of cleansing with the gauge valve closed. Then gradually open the gauge valve ufficiently for the operator to be able to read a steady pressure upon the gauge.

If the gauge continues to show a reading considerably above the correct atomising release pressure, further pumping should be done with the gauge valve closed, the aim being to clean the injector in this way by the force pump until a further trial shows the gauge to read approximately the correct release pressure, as stamped on the plate attached to the injector.

A good spray should cut off without a dribble at each pump stroke and the quantity of spray from each hole should be approximately the same.

If it is necessary to dismantle the injector, the instructions given in this espect by the individual makers should be closely followed.

Re-conditioning Equipment

The use of this reconditioning equipment is fully explained by the suppliers. The instructions are simple and straightforward, and the possession of such a kit is to be recommended to every user of a diesel locomotive.

It is emphasised that the preceding instructions (for which we are indebted o Messrs. Ruston & Hornsby, Ltd.) should only be regarded as a guide, and more specific information concerning any particular type of locomotive can be obtained from the operating and instruction manuals supplied by the nanufacturer.

TESTS ON DIESEL-TYPE ENGINES

NOW that compression-ignition engines are being used for motor vehicles, and are being made to run at very high speeds, it is important, so far as indicators are concerned, to distinguish between engines of this type and those running at comparatively slow speeds, such as ships' main engines and auxiliaries, and engines used for driving large electric generators. Each type has its own indicator, and the following description relates to the medium-speed model. We shall consider indicating, say, a ship's main diesel engine, the apparatus required consisting of the indicator, indicator valve, and reducing gear, all of which differ from the corresponding units used for steam engines.

The Indicator

Fig. 1 shows a well-known type of diesel-engine indicator, and although it operates on a similar principle to the steam-engine indicator, it has a number of features to suit the exacting conditions met with in compression-ignition practice. Since cylinder pressures are much higher, the indicator is strengthened throughout, and while double—instead of single—coil springs are used to prevent the use of unnecessarily heavy spring wire, which would unduly stress the instrument, a piston of reduced area is fitted. This means that with the same engine pressure in the indicator cylinder, a spring of only half strength gives the same diagram height with a half-area piston and only half the force is felt by the piston rod. The piston, which is hardened and of heat-resisting steel, has grooves cut in its surface to collect carbon and other grit blown into the indicator cylinder by the engine gases. These channels also accommodate lubricating oil. The recording drum allows for a diagram 1 in., 1½ in., or 2½ in. high according to the size of indicator chosen.

A notable feature of this model is the incorporation of a cooling device so that it can be used on high-temperature tests. Spare cylinders and pistons are also provided, and a quick-acting drum-spring adjustment is fitted, together with a "detent" gear, so that the drum can be stopped for changing the diagram paper without unhooking the cord.

The Diesel Indicator Valve

When indicating a diesel engine, it is found that if an ordinary indicator cock is used, seizure of the cock plug takes place owing to the intense heat. Therefore a valve is used instead. A convenient type of valve is shown in

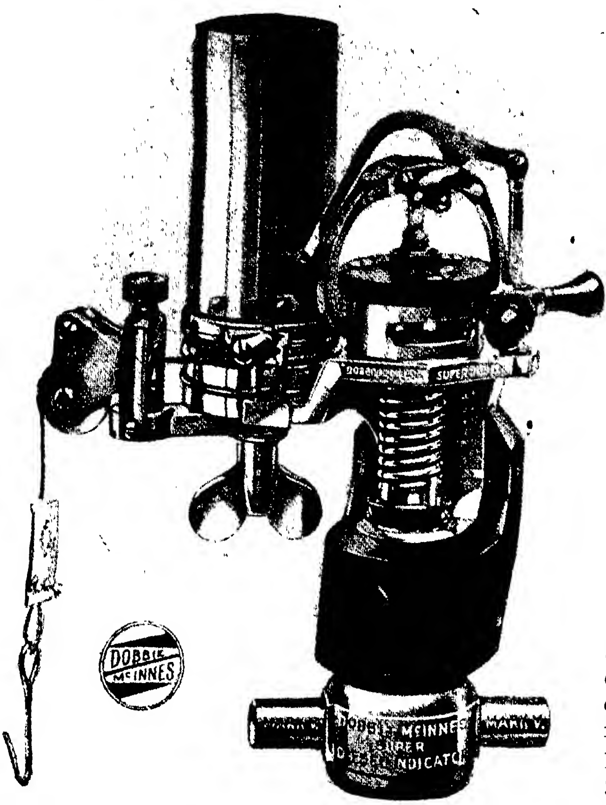


Fig. 1.—"Dobbie-McInnes" diesel-engine indicator for medium-speed engines up to 800 r.p.m.

section in Fig. 2. No stuffing box is required with this valve, and as the spindle is screwed with a "quick" thread, the full movement can be accomplished in under two turns of the handle.

Reducing Gears

The camshaft is conveniently used for the reducing gears which actuate each indicator drum, so that the movement of the latter shall be a reduced-scale copy of the motion of the particular engine piston. Before the gears are fitted, it must be remembered that the drums must be pulled forward and backward once while the respective pistons move from top centre to bottom centre and back again; also that the two-cycle engine camshaft rotates once for every revolution of the crankshaft and the four-cycle camshaft only half a turn per crankshaft revolution.

Fig. 3 illustrates an eccentric-type gear often fitted to the camshaft of a two-cycle engine; it will be noted that it is really a small reproduction of the connecting rod and crank of the engine, and to obtain a correct diagram it is important that:

$$\frac{\text{distance } a}{\text{distance } b} = \frac{\text{length of engine connecting rod}}{\text{length of engine crank}}$$

For a four-cycle engine a double-cam gear is frequently used (see Fig. 4).

When constructing this type of gear, it is insufficient merely to fit two semi-circular cam profiles on opposite sides of the shaft, even though they may be accurately set so that their peaks are under the rocker-arm roller when the engine is on top dead centre; the cam profiles must be accurately designed to allow for what is known as the eccentricity of the connecting rod. No allowance need be made for this when fitting a reducing gear to an engine crosshead, but we are now dealing with a rotary part between which and the piston the connecting rod and crank intervene.

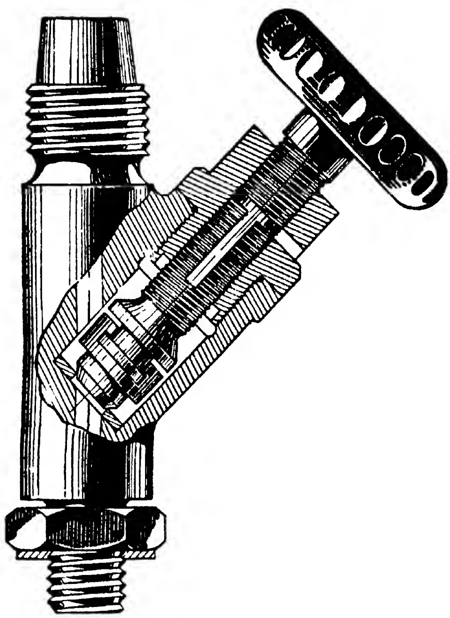


FIG. 2.—"DOBBIE-McINNES" DIESEL INDICATOR VALVE, SHOWN CLOSED TO ENGINE AND INDICATOR, OPENED TO ATMOSPHERE

Connecting Up the Reducing Gear

If reducing gears have not been fitted by the engine builders, this must be done in accordance with the above, and the gears must be set so that the points to which the indicator leads are to be attached will move in phase with the respective pistons. Indicators and valves are then screwed into the cylinder tail pipes, which should be of large bore and as straight as possible. A lead made of indicator cord, steel tape, or wire is stretched between the hook or ring on each gear by a strong spring to a fixed pin on the engine, the spring being used to keep the lead taut—see Fig. 5—and the lead should be taken as near to the indicator as possible, with the pin preferably beyond it. A loop is made in the lead or attached to it for the indicator cord hook.

Setting the Drum Cord

Any one piston is put on top centre, at which point the indicator lead will be at the end of its stroke. The drum cord is lengthened or shortened until, when hooked to the loop, the drum is clear of the stop. On slowly turning the engine one revolution, the drum should rotate forwards and backwards without reaching either stop.

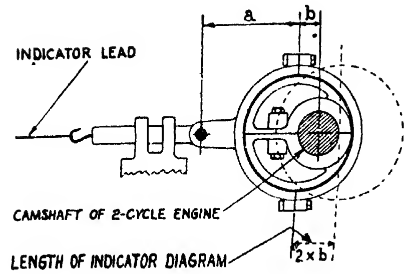


FIG. 3.—ECCENTRIC-TYPE GEAR

a/b must equal engine connecting rod to crank ratio.

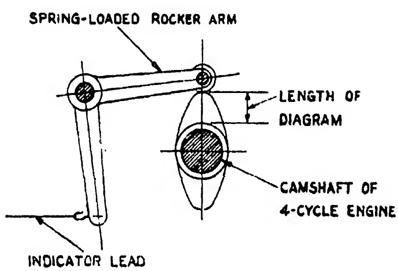


FIG. 4.—CAM-TYPE GEAR

Cam profiles must be accurately designed. Rocker arms shown of equal length.

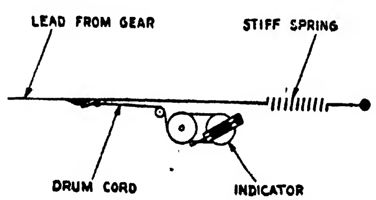


FIG. 5.—METHOD OF CONNECTING INDICATOR

Stiff spring relieves indicator drum spring from necessity of keeping lead taut.

Indicator Spring

After cleaning and oiling the indicator piston, a pressure spring is chosen for the instrument, from a knowledge of the maximum engine pressure, to give the requisite diagram height. On re-assembly, the instrument is ready for the test, the drum cord being *unhooked, the valve shut to the engine, and a diagram card placed on the drum.*

The Test

When engine conditions are reached for which indicator diagrams are required, the drum cord is hooked to the loop to set the drum in motion. The valve is opened to the indicator, the pencil of which is lightly put in contact with the paper for one cycle and withdrawn. The valve is shut and the pencil is again applied to draw the atmospheric line. Since there is no condensation to clear away, as in the steam engine, the valve should always be shut, except when actually taking the card.

DIESEL-ENGINE DIAGRAMS

Five important types are shown in Figs. 6–10, which are copies of actual diagrams obtained from large six- and eight-cylinder marine engines.

Four-cycle Engine P,V. Power Diagram

Fig. 6 illustrates the type of diagram obtained as described above. It will be noticed that at the peak, where firing takes place, the diagram is very narrow, because the indicator drum is at the end of its travel and is moving very slowly. To investigate what is happening during combustion a draw diagram is taken; the most important part of this is shown on the right of

the pressure-volume diagram, and is obtained by pulling the indicator drum cord by hand as the pencil rises and falls. The point at which firing begins is clearly shown, and the height of the outline at this point gives the compression pressure. *AL* is the atmospheric line common to both diagrams.

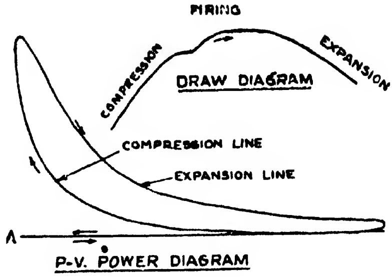


FIG. 6.—FOUR-CYCLE ENGINE INDICATOR DIAGRAM WITH HAND-OPERATED DRAW DIAGRAM ON SAME CARD

Four-cycle Engine Light-spring Diagram

To examine events during exhaust and intake, it is necessary to magnify the bottom of the diagram which appears in Fig. 6 as straight lines coincident with the atmospheric line. A light spring is therefore fitted to the indicator and the result is shown in Fig. 7.

Crank-angle Base Diagram

Since the shape of the peak of a diesel diagram is of such importance, means are often provided to give a mechanically operated draw card, so that the diagram has a form similar to the draw diagram shown in Fig. 6, but can be calibrated horizontally as well as vertically.

Compression Diagram

This is obtained by shutting off the fuel from the cylinder being indicated and is used to test the setting and accuracy of the indicator gear. Its correct shape is as shown in Fig. 8, with compression and expansion lines apparently coincident. Should it be looped, the gear requires adjustment.

Continuous Diagrams

It is sometimes of value to study changes in the form of the diagram under varying conditions. For this purpose the continuous-diagram indicator is used, and gives a complete record of consecutive diagrams on the same paper. A roll of paper is used, which is wound from a spindle inside the drum round the periphery of the drum and back to a second internal spindle. Movement of the paper with respect to the drum occurs automatically during the intake or exhaust stroke, and therefore does not interfere with the form of the diagram, which is the normal p.v. diagram repeated. Fig. 9 shows

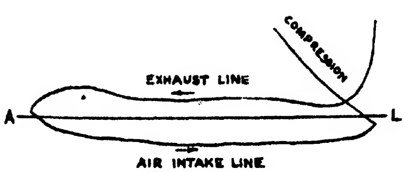


FIG. 7.—FOUR-CYCLE ENGINE LIGHT-SPRING DIAGRAM MAGNIFYING THE EVENTS DURING THE TWO "IDLE" STROKES

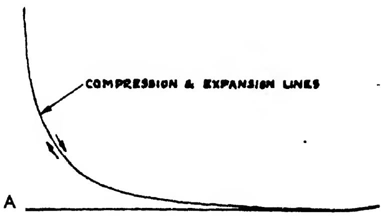


FIG. 8.—COMPRESSION DIAGRAM TAKEN WITH FUEL SHUT-OFF TO CYLINDER BEING INDICATED

continuous diagrams from a four-cycle diesel engine running at full load. Fig. 10 illustrates the pressure and power changes while the engine starts up and stops. It should be noted that Figs. 9 and 10 are tracings of the originals, and that in Fig. 10 the toes of diagrams 2–7 have been omitted, as in this particular test only the peaks were under consideration.

Engine Faults

To obtain maximum economy in running, and to ensure there are no undue strains on the engine, it is essential that valves should open and close at the correct points of the cycle and that combustion should be even. Valve setting is checked by examination of the indicator diagram, which also shows such faults as choked atomisers, early or late firing, over- and under-loading of the engine. Fig. 11 shows an example of late firing. Note the dip at the top of the draw card and the low maximum pressure as compared with the compression pressure; such conditions prevent the particular cylinder from giving full power and efficiency.

MEASUREMENT OF M.I.P. AND I.H.P.

The mean indicated pressure, sometimes called indicated mean effective pressure—I.M.E.P.—can be found by the planimeter, an instrument for measuring areas, or by the following simple method:

Referring to Fig. 12, draw a straight line perpendicular to the atmospheric line AL at each end of the diagram AB and LM, and divide the distance between the perpendiculars into 10 equal parts, A-1, 1-2, 2-3, etc. At the midpoint of each division draw 10 straight lines, *aa*, *bb*, *cc*, etc., also perpendicular to the atmospheric line. Find the total length of those parts of *aa*, *bb*, *cc*, etc., contained by the diagram, multiply by the pressure scale of the diagram, and, by dividing by 10, average the result to give the required mean pressure. For example, if

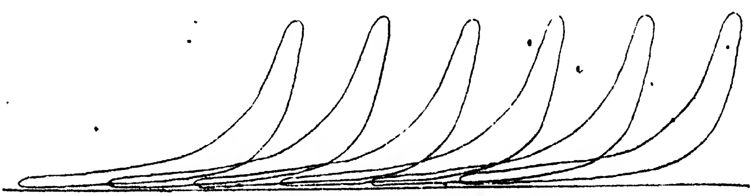


FIG. 9.—DIESEL-ENGINE DIAGRAMS FROM CONTINUOUS-DIAGRAM INDICATOR

Horizontal line at bottom is drawn by a second pencil, and may be made coincident with atmospheric line as shown in Fig. 10, Engine running on full load,

the total length of the "mean ordinates" was found to be 2·50 in., and the indicator spring scale was 360 lb./sq. in. per inch, then:

$$\text{M.I.P.} = \frac{2{\cdot}50 \times 360}{10} = 90 \text{ lb./sq. in.}$$

I.H.P. is obtained from the M.I.P. thus found, by multiplying it by the product of the stroke in *feet L*, the cylinder area in *square inches A*, and the number of *working strokes* per minute N, dividing the result by 33,000, i.e.:

$$\text{I.H.P.} = \frac{\text{M.I.P.} \times L \times A \times N}{33{,}000} \text{ per cylinder.}$$

For a two-cycle engine N is the same as the number of revolutions per

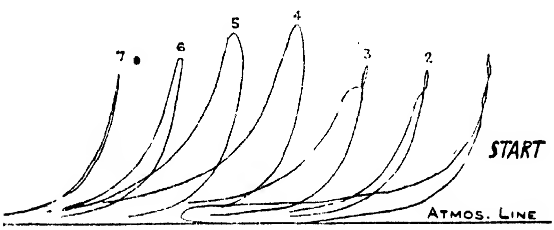


FIG. 10.—DIESEL-ENGINE DIAGRAMS FROM CONTINUOUS-DIAGRAM INDICATOR, TAKEN DURING STARTING AND STOPPING

1. Air impulse (pressure carried nearly full length of stroke). 2. Firing lightly. 3. Firing (ignition late). 4. Firing heavily (handles hard over). 5. Full power (fuel being shut off). 6. Fuel being shut off. 7. Fuel off (almost a compression card—engine nearly stopped).

minute; for a four-cycle engine N is half the r.p.m., as there is only one working stroke every two revolutions.

Use of the Planimeter

The above method of determining M.I.P., while frequently used, is not as accurate theoretically as one which enables the diagram area to be determined, thus eliminating calculation by mid-ordinates. The "Amsler" No. 6 planimeter shown in Figs. 13 and 14 measures the actual area of the diagram, and it has, in addition, an arrangement for finding the mean height.

Thus, a small variation at the peak of the card, which would not be taken into account by the mid-ordinate method, would be shown in the result obtained by the planimeter.

The diagram card is pinned to a drawing-board, and, as shown in Fig. 13, the planimeter is reversed.

By moving the slide on the tracing arm, the planimeter is set so that the distance between the points on the upper side of the arm is equal to the width

of the diagram. Without altering this distance, the planimeter, Fig. 14, is placed on the board in a convenient position with the needle point *E* outside the diagram and the pointer *F* resting on the outline of the diagram. An initial reading is taken on the dials *G* and *D*. Without disturbing *E*, the pointer *F* is carefully traced round the diagram, following every feature of the curve until one complete circuit is made. The final reading is taken, and the initial reading subtracted from it. Dividing the result by 0·4 gives the mean height in inches, which, when multiplied by the scale of the indicator spring, gives the M.I.P. from which I.H.P. is found in the usual way.

Example :

Second reading of planimeter	1·784
First reading of planimeter .	1·682
Difference .	0·102

Divide by 0·4 = 0·255 in. mean height.

If scale of spring is 360 lb./sq. in. per inch:

M.I.P. = 0·255 × 360
= 91·8 lb./sq. in.

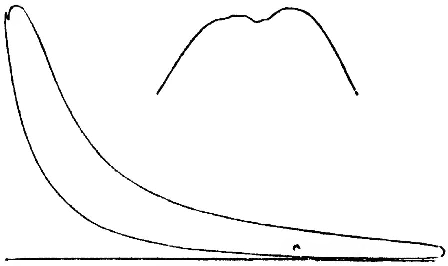


FIG. 11.—DIAGRAM SHOWING LATE FUEL INJECTION
Compare with Fig. 6.

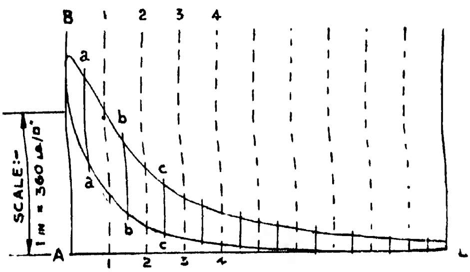


FIG. 12.—THE "MEAN ORDINATE" METHOD OF MEASURING UP THE DIAGRAM FOR CALCULATION OF M.I.P. AND I.H.P.
(M.I.P. of this diagram is 90 lb. per square inch.)

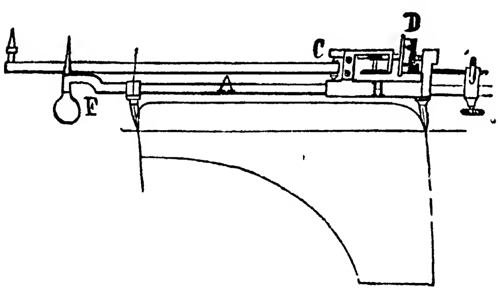


FIG. 13.—SETTING THE NO. 6 PLANIMETER TO THE DIAGRAM WIDTH

It is now an easy matter to calculate the indicated horse-power of the engine by using the formula

$$\text{I.H.P.} = \frac{\text{M.I.P.} \times L \times A \times N}{33{,}000},$$

bearing in mind that the formula gives the horse-power of one cylinder only. The total I.H.P. of a four-cylinder engine, for example, would be four times the value given by the formula.

The Optical Indicator

This type of indicator employs, instead of a link-work and pencil mechanism, a small mirror which is capable of being deflected in a vertical plane by the pressure existing in the engine cylinder, and in a horizontal plane by the to-and-fro movement of the piston in the cylinder. A beam of light focused on the mirror is reflected back from it, and will trace out an indicator diagram on a suitable screen or photographic plate. These indicators are chiefly used for laboratory and research work.

The "Farnboro" Indicator

This is an electrical indicator in which the varying pressure in the cylinder of the engine under test is recorded by means of electric sparks which puncture the indicator card.

As only very slight movements of a pressure disc or diaphragm are required to cause the sparks, this indicator is almost free from inertia effects. It is particularly adapted for use on high-speed engines, and has been used largely in the testing of motor car and aero engines.

Construction and Operation of the "Farnboro" Indicator

The essential parts of the "Farnboro" indicator are illustrated in Fig. 15. The components are as follows:

An induction coil for producing high-tension sparks.

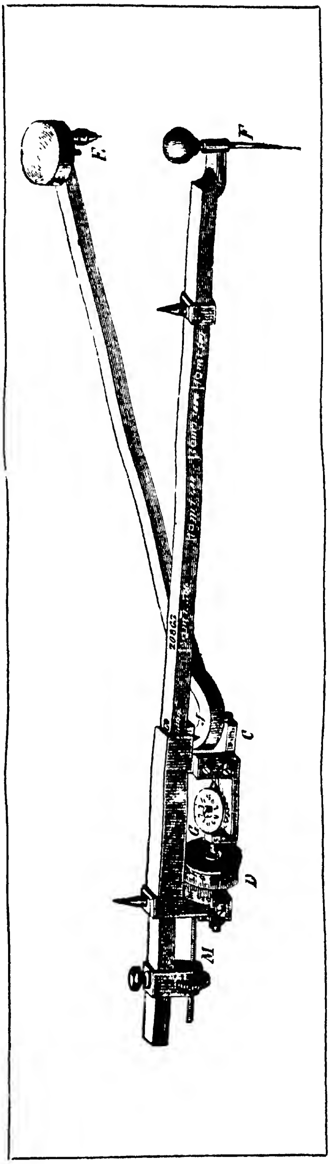


FIG. 14.—"DOBBIE-MCINNES" "AMSLER" NO. 6 PLANIMETER FOR MEASURING M.I.P. AND I.H.P. INDICATOR DIAGRAMS

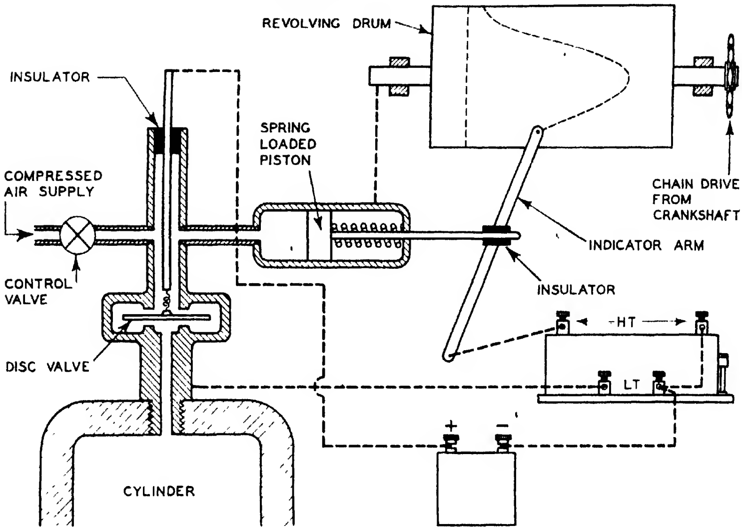


FIG. 15.—THE "FARNBORO" INDICATOR

A pressure contact unit which is screwed into the cylinder head of the engine to be tested.

A compressed-air container with control valve.

A rotating drum for holding the indicator card.

A spring-controlled pivoted recording lever.

It will be seen that the high-tension winding of the spark coil is connected to the recording lever, the high-tension circuit being completed through the metal drum and the engine frame.

The low-tension side of the ignition coil is connected to a movable metal disc inside the pressure unit. This disc, which can move between two seats, interrupts the primary circuit of the coil whenever it leaves either the upper or the lower seat.

The air container is connected through its control valve with the upper side of the pressure unit, so that when the control valve is opened slightly the disc is forced on to its lower seat and the air pressure is transmitted to a spring-controlled piston which controls the movement of the recording lever. The position of the sparking point along the axis of the drum is determined by the air pressure acting upon the upper side of the pressure unit. The rotating drum is driven from the engine crankshaft directly or by chain, so that the angular position of the drum corresponds to the angular position of the crank.

When the pressure in the engine cylinder rises above the value of the air pressure, the disc valve lifts from its seat and a spark occurs between the

recording lever and the drum. Similarly, when the cylinder pressure falls below the air pressure, the disc leaves the upper seat and a spark occurs. These sparks mark the indicator card.

To obtain an indicator diagram, the engine should be run up to a speed, and when steady conditions have been reached an increasing pressure is admitted from the air reservoir to the upper part of the pressure unit. As the air pressure rises, the spring-controlled piston in the pressure unit gradually moves outwards, carrying with it the recording lever.

The disc valve leaves the lower seat every time the engine cylinder pressure exceeds the air pressure above the disc. In the meantime, the indicator drum is rotating in synchronism with the crankshaft. In this way an indicator diagram is obtained in the form of spark punctures on the indicator card. The line joining these punctures gives a pressure record plotted against crank angles. This may afterwards be converted to obtain an indicator diagram showing cylinder pressures plotted against piston displacements.

The Cathode-ray Indicator

This is one of the latest and most convenient types of indicators for use with compression-ignition engines. The indicator unit consists of a small metallic capsule which can be screwed into the cylinder head, as shown in Fig. 16. When the varying cylinder pressure acts on this capsule, electrical voltages are generated in it, the voltage being proportional to the pressure. Two electrical leads from this unit are taken to a cathode-ray indicator. The horizontal sweep of the cathode-ray beam is controlled by an electro-mechanical arrangement operated from the engine shaft, so that

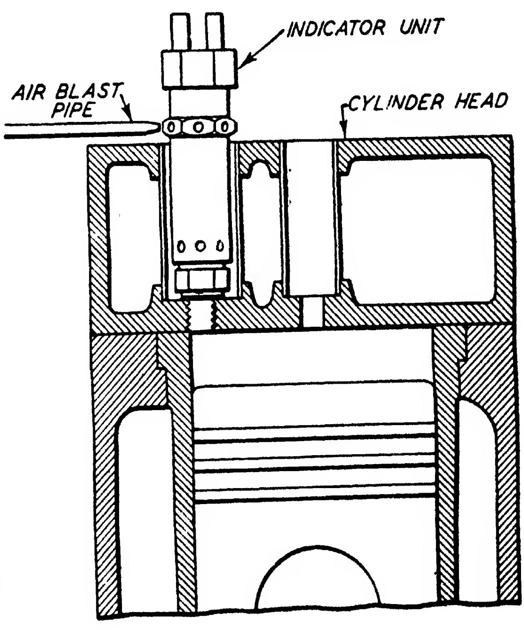


Fig. 16.—Cathode-ray indicator unit fitted to cylinder

The indicator unit is cooled by directing a blast of compressed air at one of the ventilating holes in the bay.

the spot moves horizontally across the screen in synchronism with the movement of the piston in the engine cylinder.

The varying voltages generated in the indicator unit control the vertical deflection of the cathode ray, so that when the apparatus is in use an indicator diagram can be seen on the screen of the cathode-ray tube. With such apparatus means are provided for obtaining a permanent record on a photographic plate whenever this is required.

MEASUREMENT OF B.H.P.

For testing purposes it is, of course, necessary to provide an artificial load for the engine. There are four possible methods of doing this, namely, by the use of a rope or band brake applied to the rim of the flywheel; by means of an electric dynamometer; or by the use of an air brake.

For small- and medium-sized engines some form of friction brake is the simplest. This method also requires less elaborate equipment than the dynamometer methods referred to below.

Where a large number of tests are to be carried out on a range of engines, e.g. in the production testing, a dynamometer load is the most suitable. For general-purpose work the Froude hydraulic dynamometer is most satisfactory. For very accurate type tests some form of electric dynamometer is probably the most suitable.

Fo rvery small high-speed engines an air brake offers the advantage of great simplicity in use, though it involves special care being taken in observing atmospheric conditions, as the resistance offered by the air screw varies according to the barometric pressure, the air temperature, and the humidity during the period of the test.

Friction Brakes

The two main types of friction brakes used are the rope brake and the band brake.

The arrangement for applying a rope brake to an engine which is about to undergo a test is shown in Fig. 17. A flanged rim is attached to the flywheel, and a rope wrapped once round. Weights are attached to one end of the rope and a spring balance supports the other end. As the engine flywheel rotates, the friction between the rope and the rim tends to lift the weights, so that the spring balance indicates a figure less than the total weight on the brake. The difference between these weights, multiplied by the "lever arm" (see Fig. 17), gives the torque on the engine.

It will be noticed that the rim, which is bolted on to the flywheel, has an outer flange. It will also be seen that there are two pipes, the ends of which are bent over this flange so that they are located near the inner rim of the brake attachment. A supply of water passes from one of these pipes into the interior of the brake rim when the latter is revolving, and this water is scooped up by the outlet pipe after it has passed round the wheel. This prevents the friction of the rope from overheating the brake rim.

The points requiring special attention when using the rope brake are as follows :

1. The diameter of the rope must be taken into account when calculating the lever arm, as shown in Fig. 17.

2. The weight of the rope between the points *XX* and *YY* should be allowed for. *XX* should be taken from the weight shown on the balance, and *YY* added to the main weight.

3. A restraining bracket should always be fitted to prevent the weights from being thrown over the wheel should the rope suddenly "seize" the brake rim.

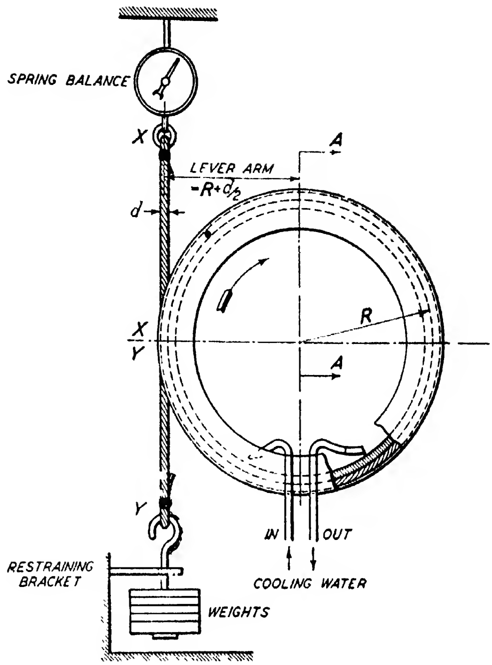


Fig. 17.—Arrangement of the rope brake

The limitation of the rope brake is that the friction between the rope and the wheel is liable to fluctuate slightly. These fluctuations will, of course, be seen on the spring balance, but an intelligent observation of this will enable fairly correct average reading to be obtained.

If a weight of 50 lb. is placed on the hook at the lower end of the rope, the spring balance should register a little over 50 lb., because it should also show the weight of the hook supporting the weights and also the weight of the straight part of the rope. If the engine is now started, with the rope wrapped round the brake rim, the rope will tend to be carried round in the direction of rotation. This will have the effect of decreasing very considerably the weight indicated on the spring balance.

Assuming with the engine at rest the spring balance gives a reading of, say, 52 lb., and with the engine rotating over a certain period the average reading is 27 lb., then the braking force is equal to 25 lb., acting at a leverage equal to the distance from the centre of the flywheel to the vertical centre line of the rope.

To determine the horse-power the engine is developing under load, it is necessary to observe the speed of the engine shaft in revolutions per minute by means of an engine counter or by means of a speed indicator.

The resolution counter requires an observation to be taken over a period of a minute or more. The results obtained are highly accurate, but they represent the average speed over the period of observation.

The speed indicator gives an instantaneous reading of the speed. It also shows whether the speed is steady or fluctuating, but from the point of view of accuracy it is not so reliable as the revolution counter, because in course of time the control springs of the governor mechanism, or the field strength of the generator magnets, may become weakened. Such an instrument, therefore, needs fairly frequent recalibration if accurate results are required.

B.H.P. Calculation

If in an engine undergoing a friction brake test the following readings are obtained—

Brake load, 70 lb.
Effective radius of brake, 24 in.
Speed, 250 revs. per min.,

the brake horse-power can be calculated as follows :

Force applied to rim of brake, 70 lb.
Distance moved by rim of brake in 1 min.

$$= \frac{250 \times 24 \times \pi 2 \text{ ft.}}{12}$$

$$= 3{,}142 \text{ ft.}$$

$$\text{Work done per min.} = 70 \times 3{,}142 \text{ ft./lb.}$$

$$1 \text{ h.p.} = 33{,}000 \text{ ft./lb. per min.}$$

$$= \frac{70 \times 3{,}142}{33{,}000}$$

$$\text{Brake horse-power of engine under test} = 6{\cdot}66 \text{ horse-power.}$$

The Band Brake

This is an alternative to the rope brake, the rope being replaced by a steel band to which wood or fibre blocks are attached (see Fig. 18). This band can be tightened on to the wheel to increase the load by means of a screw. A lever of fixed length is attached to the band, and the torque is measured on a spring balance. A balance weight is fitted to balance the weight of the lever, or the weight of the lever can be subtracted from the spring balance reading (a sight feed is most satisfactory), and the rim must be cooled. Many engineers admit a trickle of water to the cooling channel, allowing the water to boil, but a more satisfactory method is to fit a scoop to withdraw the water.

The chief difference in the method of calculating brake horse-power when using the band brake as compared with the rope brake is that instead of using

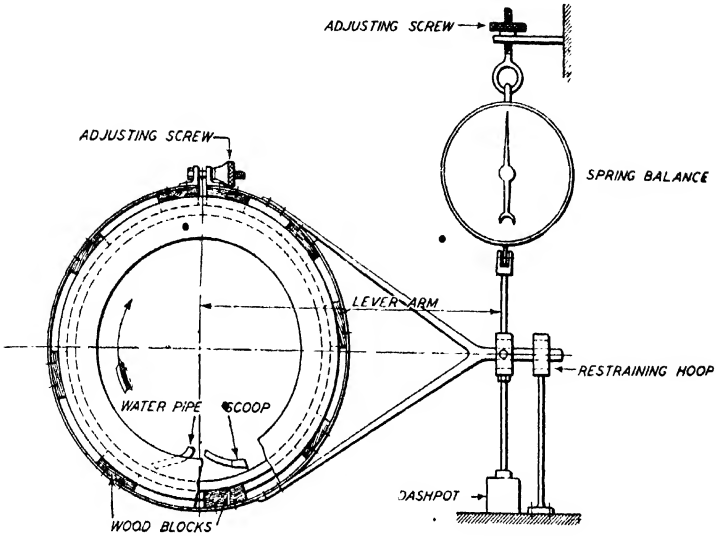


FIG 18.—THE BAND BRAKE
This type of friction brake is more accurate and stable than the rope brake.

the effective radius of the brake rim as the lever arm the measurement must be as shown in Fig. 19, i.e. the effective leverage must be measured from the centre of the flywheel to the centre line of the spring balance.

The Froude Dynamometer

The Froude dynamometer is a device which is suitable for coupling up to an engine to be tested. It provides an artificial load, utilising the principle of hydraulic friction. A detailed description is as follows:

The main shaft carries a rotor which revolves inside a watertight casing (see Fig. 20).

Each face of the rotor is formed into a series of semi-elliptical pockets, divided from each other by a series of oblique vanes. The internal faces of the casing are pocketed in the same way, but these pockets face in the opposite direction to those in the rotor. Between the rotor and the casing are fitted adjustable sluice plates, and a supply of water is fed into the pockets through holes in the casing.

As the rotor revolves, the water is flung outwards and forwards out of the pockets. The streams of water so formed are reversed by the casing pockets and flung back into the rotor. The reversing of the flow in this way exerts a force on the casing, and tends to drag it round with the rotor. The casing is mounted on rollers, and is restrained from movement by a weighing gear (spring balance

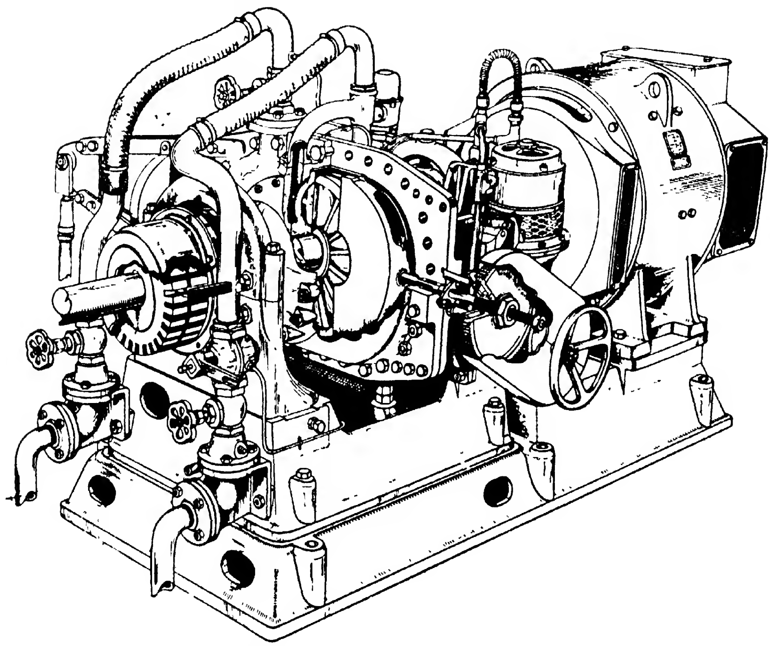


FIG. 19.—SECTION OF FROUDE HYDRAULIC DYNAMOMETER

The electric motor is for driving the engine to determine pumping losses, and is declutched when not required. It may also be employed as a starting motor.

or weights, or both). Measurement of the restraining force required, which includes, not only the hydraulic reaction, but also the gland and bearing friction, gives a measure of the power output of the engine. The load is adjusted by operating the sluice plates, which blank off more or less of the easing cups from the water stream.

For calculating the brake horse-power of an engine when using the Froude dynamometer, the makers of this equipment provide a simplified formula which takes account of the dimensions and characteristics of the particular model used. For example, a typical formula is:

$$\text{Brake horse-power} = \frac{WN}{1{,}000},$$

where W is the weight shown on the spring balance and N is the speed in revolutions per minute. The factor 1,000 would, of course, vary according to the particular model of dynamometer in use.

The Froude dynamometer is very suitable for routine and production

testing. It is very accurate and easy to use, providing the following simple precautions are taken:

The brake should always be carefully balanced, so that the weighing gear reads zero with the rotor stationary, but with water running through the brake.

The brake arm must be exactly level when taking readings, and a spirit level may be attached to the brake arm to ensure this.

The brake should be examined from time to time to see that there are no flats on either the supporting rollers or on the race.

The water supply to the brake should be from a constant level header tank, and if more than one brake is run from the same tank the supply main should be very much larger than normal practice for water mains, and should be as short and as free from bends as possible.

The Electric Dynamometer

This device is similar to the hydraulic or Froude dynamometer in that it is a separate attachment to be coupled up to the engine under test, but instead of providing a mechanical load it provides an electrical load for the engine.

The usual type employs a D.C. generator which has been carefully calibrated so that its efficiency at all loads and speeds is known. The power generated, instead of being wasted, is sometimes fed into the works supply and a continuous record is kept throughout the test of the amount of electrical energy generated.

The Torque Reaction Dynamometer

This is another form of electric dynamometer in which the generator casing is mounted on trunnions in a similar manner to the casing of the hydraulic dynamometer. A spring balance or weighing gear is used to measure the torque reaction, and the calculations are made as for the Froude dynamometer.

The torque reaction dynamometer is more accurate than the electric type, and is independent of the varying efficiency of the electrical generator at different loads. It does not require continuous observation or recording of the electrical power generated.

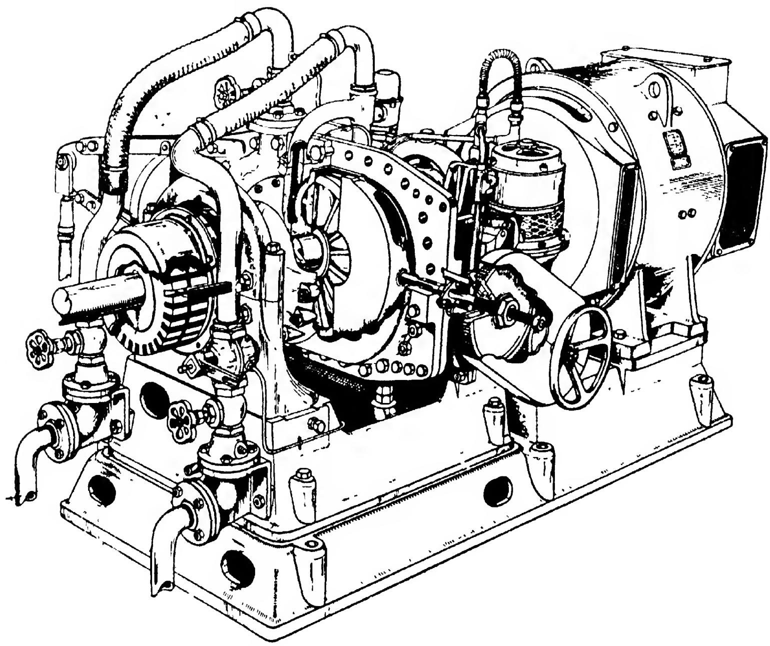


FIG. 19.—SECTION OF FROUDE HYDRAULIC DYNAMOMETER

The electric motor is for driving the engine to determine pumping losses, and is declutched when not required. It may also be employed as a starting motor.

or weights, or both). Measurement of the restraining force required, which includes, not only the hydraulic reaction, but also the gland and bearing friction, gives a measure of the power output of the engine. The load is adjusted by operating the sluice plates, which blank off more or less of the easing cups from the water stream.

For calculating the brake horse-power of an engine when using the Froude dynamometer, the makers of this equipment provide a simplified formula which takes account of the dimensions and characteristics of the particular model used. For example, a typical formula is:

$$\text{Brake horse-power} = \frac{WN}{1{,}000},$$

where W is the weight shown on the spring balance and N is the speed in revolutions per minute. The factor 1,000 would, of course, vary according to the particular model of dynamometer in use.

The Froude dynamometer is very suitable for routine and production

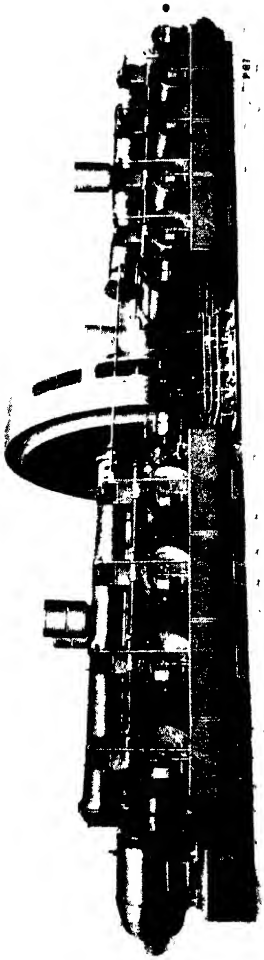


FIG. 1.—CROSSLEY-PREMIER 2,000-B.H.P. VIS-À-VIS GAS ENGINE

(Crossley-Premier Engines, Ltd.)

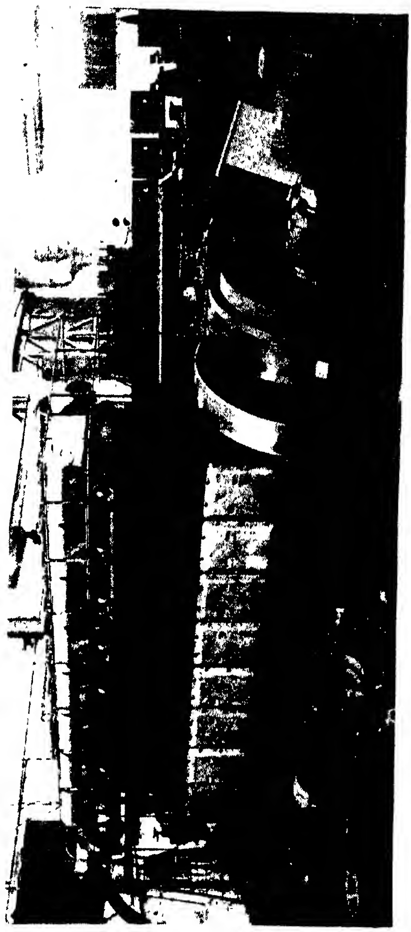


FIG. 2.—NATIONAL VERTICAL DUAL-FUEL ENGINE ON WORKS TEST PLATE

Output, 1,332 B.H.P. on oil and 1,000 B.H.P. on natural, town's, or sewage gas. Speed, 333 r.p.m. *(National Gas and Oil Engine Co., Ltd.)*

tandem engines (chiefly for use in ironworks), developing 5,500 B.H.P. or more. The usual fuels for these large-size engines are blast-furnace, natural, or sewage-sludge gas. Smaller engines may be run additionally on producer or town's gas, and nearly all sizes of engines are available for operation on coke-oven gas. Speeds range from 1,000 r.p.m. to 100 r.p.m. or slower.

The vertical form of conventional engine is available for operation on any gaseous fuel in powers ranging from 5 B.H.P. up to 3,500 B.H.P. or thereabouts, the speeds being comparable to those of oil engines with the same output. (In fact, the vertical gas engine is almost indistinguishable from the oil engine except for its greater silence in running.) Mention must also be made of the Nordberg radial engine, which has the crankshaft axis vertical. The engine is mounted on top of the electric generator in the same way as in the case of a water turbine. Considerable economy of floor space is claimed, and the fact that there is a single-throw crank reduces the problem of torsional vibrations very considerably. The brake mean effective pressure is also often higher.

The High-compression Gas Engine

The high-compression gas engine is, at present, only available in vertical form, having cylinder dimensions of less than 6 in. bore and 9 in. stroke. Larger sizes have been built experimentally, but trouble, due to detonation at the high-compression ratio, is experienced if these sizes are much exceeded. Thus, the upper power limit, with eight vertical cylinders, is about 200 B.H.P. at 1,000 r.p.m.

The average full-load fuel consumption of the conventional gas engine is about 9,500 B.Th.U. per B.H.P.-hour, and is little affected by the type of gas used. For the high-compression type a fuel consumption of about 7,200 B.Th.U. per B.H.P.-hour should be expected, and, in favourable conditions, figures as low as 7,000 B.Th.U. per B.H.P.-hour have been returned. Thus, the high-compression engine shows a fuel saving of about 25 per cent. compared with the conventional type.

The Dual-fuel Engine

The dual-fuel type is available in sizes from 10 B.H.P. at 1,500 r.p.m. up to about 1,000 B.H.P. at 333 r.p.m., with normal aspiration, and from 1,500 B.H.P. up to about 7,000 B.H.P. as gas-injection engines (see pages 95 to 97). The latter are suitable only for natural gas of high calorific value. These engines are all vertical. The fuel consumption of the dual-fuel engine is usually slightly better than that of the high-compression type, possibly due to the better ignition of the charge, and figures as low as 6,700 B.Th.U. per B.H.P.-hour (including oil and gas) have been obtained.

The Instantly Convertible Gas Engine

The instantly convertible oil-gas engine is, in effect, a high-compression gas engine having magneto ignition, with the addition of the necessary fuel pumps and control gear to enable it to run as an oil engine. It is available in the same

sizes as the high-compression engine, and its performance on gas is the same. As an oil engine, the fuel consumption is usually better than 0·38 lb. per B.H.P.-hour. (The same figure may be expected from the dual-fuel engine running as an oil engine.)

Lubricating-oil Consumption

Lubricating-oil consumption on all types should be better than 3,000 rated B.H.P.-hours per gallon, but naturally much will depend on the speed and load factor. Both conventional and high-compression gas engines on light load tend to draw lubricating oil into the combustion space during the suction stroke; this is not the case with dual-fuel engines. The larger sizes, having metered cylinder lubrication, should return rather better lubricating-oil consumption figures. Maintenance charges are generally lower than for oil engines, but if a producer plant is required, the cost of running and attending this must be added to the fuel cost in assessing the cost efficiency of the station as a whole.

SPARK-IGNITION ENGINES

Botn high-compression and conventional engines operate on what is loosely called the "Otto" cycle, though the two-stroke cycle is also used. Gas and air are drawn into the cylinder in the "correct' mixture ratio, and compressed. Ignition by spark is followed by combustion of the charge at approximately constant volume. During the subsequent expansion stroke work is done, and the products of combustion are finally expelled so that the cycle can be repeated. Supercharging (or more properly, pressure-charging) is frequently adopted. The efficiency of such a cycle, all other things being equal, depends mainly on the compression or expansion ratio. Moreover, as the temperature of the exhaust gases is lessened with increase in expansion ratio, so the volumetric efficiency is improved, and this, with other factors, enables a higher output to be achieved as the compression is raised. (For a given cylinder, an increase in mean effective pressure from 70 lb. per square inch at 6 to 1 to as much as 95 lb. per square inch at 13½ to 1 should be possible on town's gas.)

Governing Systems

Unlike petrol engines, gas engines give their best performance when the fuel-air ratio is theoretically correct. A well-governed gas engine will show practically constant exhaust temperature from about 25 per cent. of full load up to full load, with a slight rise at overload and a fall at no load. With few exceptions all engines are governed in such a way as to maintain a constant quality of mixture, but to varv the quantity taken in per cycle.

The exception is the "hit-and-miss" governing system, very common at one time, but now obsolete. In this system the engine is governed by preventing the cylinder from receiving any charge at all for certain of the strokes; as the speed falls, the governor then allows the cylinder to take in a full charge. The resulting working stroke is sufficient to keep the engine running for a few more revolutions until the speed again falls and another charge is admitted.The number of charges

per minute is varied according to the load demand. Clearly the speed regulation of such an engine will be very poor, and quite unsuitable for electric generation. The system is, however, thermally very efficient, owing to the very good scavenging of the cylinder between working strokes.

Two main systems of governing by quantity are in use:

CROSSLEY SYSTEM.—In this system the lift of the inlet valve is varied by an ingenious system based on a variable fulcrum lever. The inlet valve has seats controlling both air and gas, and the proportioning of these seats is designed to ensure mixture strength as near as possible correct to all openings of the valve. Adequate mixing of the air and gas, so essential at light loads, is ensured by the action of the valve seats at small openings.

NATIONAL SYSTEM.—The system most commonly associated with the National gas engine, though used by others, consists of a combined gas and air inlet valve, but of constant lift. Flow to this valve is controlled by a system of butterfly throttles in the inlet pipes. It is claimed that this system, whilst not so simple as the Crossley, enables individual adjustment both of mixture quantity and mixture strength to the individual cylinders of a multi-cylinder engine. Both systems are in very wide employment, and appear to give full satisfaction.

On smaller engines a Venturi-pattern gas-mixing valve, similar to a petrol carburettor of the elementary type, controlled by a simple butterfly valve as throttle, appears to be satisfactory. In fact, during the period 1939–45, many successful conversions from petrol to gas fuel were made by rigging up a gas mixer out of old pipe fittings, and controlling the resulting mixture by the existing petrol-engine throttle.

Adjusting Mixture Strength

Whatever system of governing is employed, it will be found that some provision is also made for temporary adjustment of mixture strength to allow for changes in gas quality. This is usually done by adjustment of the main gas cock at the end of the engine, the air supply being left wide open. Occasionally, however, it may be necessary to adjust the air cock as well. Engines designed to run on a number of gases at choice, e.g. producer or town's gas, are always provided with means for adjusting the mixture ratio easily.

It will be appreciated that as the mixture ratio is controlled by the ratio between two areas, whether the areas concerned are in the inlet valve or in the throttle, some control of gas pressure is essential. Engines operated on suction gas producers will always be receiving gas at less than atmospheric pressure; clearly, therefore, if such an engine were to be changed over from a suction to a Mond (pressure) producer, even though the gas quality were to remain the same, a change in valve setting would be required. Furthermore, if an engine is operated on gas supplied under pressure, however small, there is always a risk of gas leakage into the engine-room.

For this reason most manufacturers recommend the provision of a gas governor, which will serve the triple purpose of controlling the gas pressure to the figure at which the valves were originally set, as a safety device to prevent

accidental leakage when the engine is stopped, and also to protect the gas-metering device from error due to pulsating flow and damage due to backfire (though the backfire is very uncommon on the modern engine).

Ignition

Ignition of the charge is carried out in most cases by the use of a high-tension magneto and sparking-plugs. The low-tension system is employed in very large engines, and in a few small engines where the compression ratio is low. In the high-compression engine the density of the charge is such that "high-performance" magnetos and plugs may be necessary—these are analogous to the high-performance coils required on some modern cars. Coil ignition is seldom employed. Although it has many advantages, one serious objection is the fact that it requires a dynamo and battery. (It should be noted that whilst an automobile if worked very hard may do some 2,000 hours running a year, a gas engine may be called upon to operate for three times this figure.)

Provision should be made for advancing the ignition as required by the load on the engine. In the case of the National high-compression engine this is done automatically by the governor. The importance of this point is not always appreciated, but it will be found that if the ignition be kept as advanced as the engine will stand without distress, a considerable annual saving of fuel will result.

Construction

The construction of gas engines is similar to that of oil engines of the same size, and, in fact, the components are frequently interchangeable. The same cranks and connecting rods are used, so that there need be no fear that the bearings will suffer from the pressures caused by operating at a high-compression ratio. Crankcases and beds are identical, the necessary pads for carrying both gas- and oil-engine components being carried on the pattern. Starting is by compressed air in the larger sizes, by electric or air motor in the smaller engines, and by hand in the case of very small engines.

Built-in "Streamline" oil filters, water-cooled main bearings, individual cylinder liner lubricators, and positive crankcase ventilation are all features of the modern gas engine, as in the case of the contemporary oil engine. The absence of fuel-oil leaks and considerably quieter running are the only strikingly noticeable differences between a gas-engined and an oil-engined station.

THE NORMALLY ASPIRATED DUAL-FUEL ENGINE

The basic difference between this type of engine and the spark-ignition engine is in the means employed to ignite the charge. The dual-fuel engine always employs "high compression" (up to 15 to 1), and this is sufficient to ignite a small spray of oil injected into the gas-air charge. (It is not generally realised, but the self-ignition temperature of diesel oil or gas oil is considerably less than that of even the most inflammable of the gases.)

The burning spray of oil, after a slight delay, ignites the main charge of gas, and combustion proceeds as in the "straight" gas engine. The shape of the combustion chamber is, of course, of some importance, and considerable experiment is necessary to develop the correct profile of piston and cylinder head. Generally, "open"-type oil-engine combustion chambers will be found suitable for running on gas in this way, but precombustion chambers or similar types are not. The same remarks apply to "instantly convertible" engines. If anything, spark-ignition high-compression engines are even more critical as to shape of combustion chamber.

Fuel-oil Consumption

The quantity of oil required for ignition is small, and, in fact, is not sufficient to run the engine on no load, even at a reduced speed. The usual way of expressing the quantity of oil is to take the heating value of the amount of oil consumed per hour and to express that as a percentage of the heating value of the total fuel, oil and gas, consumed per hour at the rated full load of the engine.

For example, a 10-B.H.P. engine may use 59,000 B.Th.U. of gas per hour at full load, and require 0·33 lb. per hour, say 6,000 B.Th.U. of oil for ignition purposes. Thus, the total fuel per hour is 65,000 B.Th.U. at full load, and the proportion of oil is 6,000 to 65,000, say 9·23 per cent. This amount of oil is found by nearly all manufacturers to give the most economical *overall* fuel consumption, and although ignition can be initiated by much smaller quantities, the weight of oil per hour would remain constant at this figure at all loads on the engine.

Governing

As full compression is required to ensure the ignition of the pilot-oil charge, it is clear that governing by throttling both gas and air cannot be used. It is found, however, that even though the gas only be controlled, and the air valve be left full open at all loads, ignition of the very weak mixture which will result at low loads is still quite effective—probably due to the comparatively large surface of the igniting flame. At light loads, the mixture being very weak, the engine will be found rather less efficient than a high-compression gas engine having quantity governing and spark ignition. At half to full load, however, the improvement over the conventional gas engine is most marked. Misfiring at light loads is comparatively rare. There appears to be little or no objection to slight throttling of the air at light loads, provided compression temperature is maintained at a figure higher than the spontaneous ignition temperature of the fuel oil, and a partial closing of the main air cock would assist in curing any misfiring if prolonged running at light loads is necessary.

As mentioned above, the amount of oil used for ignition when aiming at maximum total economy lies between 8 and 12 per cent. If, however, there is a serious shortage of oil, this proportion may be reduced to 5 per cent., and, in extreme cases, to 3½ per cent. In the latter case some misfiring at low loads may be experienced.

On the other hand, the engine may be used as a "mixed-fuel" engine, and the proportion of oil increased to make up for, say, a temporary deficiency of gas. As the gas supply falls off, the oil supply is gradually increased by the operation of the special governor gear until, in the limit, the engine is running as an oil engine. In practice, it should be stated, the engine is best turned over to full oil as soon as the proportion of gas falls below one-half of the total fuel input to the engine.

Gas Governor Gear

In the National range of dual-fuel engines, those above 6-in. cylinder bore have inlet pipes and gas governor gear identical to that fitted on normal gas engines, with the exception that the throttle in the air passage is removed and the engine is governed by throttling the gas supply only. The change over from one gas to another, or from either gas to oil alone can, of course, be carried out without shutting down or reducing load. Smaller types of dual-fuel engine employ, as mentioned already, a simple gas-mixing valve; as before, only the gas supply is governed, and the mixture is distributed to the cylinders by a simple undivided manifold.

Fuel Pumps and Sprayers

If the engine is required to operate mainly as a gas engine, i.e. if the fuel-oil percentage is unlikely to rise above about 15 per cent., it is desirable to use special sprayer nozzles and pumps, so proportioned as to size and capacity as to give the most efficient igniting spray when passing this relatively small quantity of oil. In such a case, if an emergency arises in which gas is not available, a shut down of, perhaps, a couple of hours will be necessary in order to change the pump elements and sprayer nozzles.

When it is known that engines will be required to operate either as mixed-fuel engines or as oil engines with any frequency, then the fuel pumps and sprayers are designed for normal oil-engine operation. Under these circumstances, it may be found that the nozzles will require cleaning a little more frequently than is normal for an oil engine, but the trouble is neither serious nor universal.

GAS-INJECTION ENGINES

The low-pressure gas-injection engine, known as the "Erren" engine, is now only seldom used. It was the forerunner to the radial engine described below. The designer of this engine was concerned with the conversion of vehicle engines to gas operation, and, as such conversion can now be carried out by much simpler means, it is unlikely that it will again be used for this purpose. The only advantage which might justify the "Erren" conversion is the fact that, owing to the supercharging effect, the power output on gas is as great or greater than the output on oil or petrol.

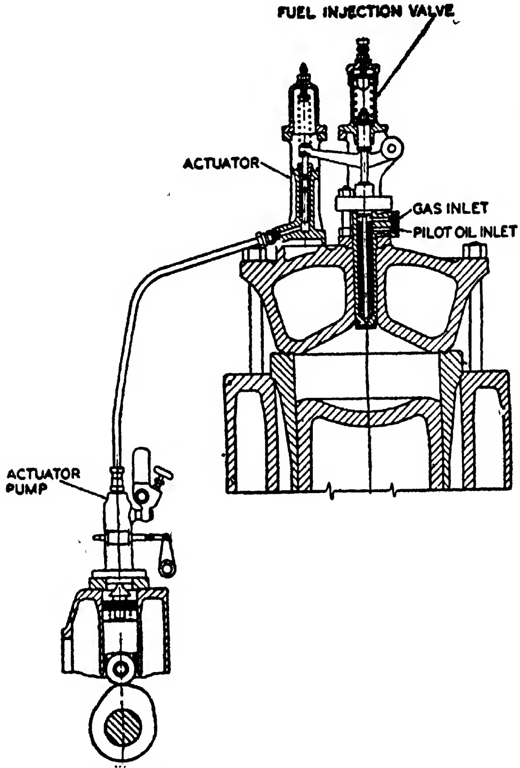


FIG. 3.—DIAGRAMMATIC ARRANGEMENT OF NORDBERG GAS-OIL INJECTION GEAR

Nordberg Radial Engine

The radial engine mentioned on page 90 is a two-stroke type, having eleven cylinders of 14 in. bore by 16 in. stroke. The output, on natural gas, is 1,125kW., say 1,650 B.H.P.; of this figure, 125kW. (100 h.p.) is alternating current required to drive the scavenge blowers, cooling water pumps, and so on. The engine is of the loop scavenge type, and gas is supplied to the cylinder at a pressure of about 6 lb. per square inch. The injection valve is mechanically operated and is mounted in a cage in the cylinder wall. The construction of the engine, apart from the lower compression ratio, is almost identical with that of the Nordberg radial oil engine.

It should be noted that this engine is a *low*-compression type with *spark* ignition, whilst the "Erren" engine was, of course, of high-compression ratio. No gas compressor is required, as the gas is obtained from the field at a pressure of 400 lb. per square inch.

The "Gas-blast" Injection Engine

This engine, developed by the Nordberg Manufacturing Company of America, was designed in the first instance to use a rich natural gas of over 1,000 B.Th.U. per cubic foot found in the Texas oilfields and elsewhere. So far it has only been built in sizes over 1,200 h.p.

It operates on the two-stroke cycle and is scavenged in the usual way. When

running as an oil engine it is a solid-injection type, using the C.A.V.-Limited injection system. To change over to gas, the engine must be shut down, the main fuel pumps disconnected from their sprayers and used as servo-pumps to operate the gas-injection valves in the cylinder heads. Smaller C.A.V.-Limited pumps are then brought into service to supply the pilot-oil for ignition.

Injection Gear

The injection valves are, in principle, similar to those employed on air-blast injection diesel engines, except that in this case the oil is injected and atomised by means of a spray of gas in place of air. Compression of the gas, to about 1,100 lb. per square inch, is effected in a compressor mounted at the free end of the engine, and here again the compressor is very similar to that used on air-blast diesel engines, and is roughly the same size. The arrangement of the injection gear is shown in Fig. 3, and it should be mentioned that earlier types required the oil pipe to the actuator to be cooled, but this has been omitted in later models.

The engine resembles, both in appearance and construction, a large marine or stationary oil engine, and the design of the working parts, as in the smaller National type, is identical with that of the oil engine.

Performance

The performance of the gas-blast injection engine is very good at light loads. This is probably due to the fact that the gas, ignition oil, and air are very intimately mixed by the injection spray, and that, in spite of the overall weakness of the mixture, when considering the cylinder as a whole, it closely approaches theoretical proportions in the zone of combustion. These engines have been installed in units of many thousands of horse-power, and have given years of satisfactory service. It will be realised that such a simple expedient of using a blast-air compressor to provide injection gas can only be employed on fuels having a high heating value, such as natural gas or pure methane. The size of compressor that would be required, to say nothing of the injector, for even town's or coke-oven gas would be such that an independent compressor would be required and some other arrangement might have to be made to accommodate the injector. The design of such an engine to use, say, blast-furnace gas is just possible on paper, but many serious problems would have to be overcome. In its own field, however, the Nordberg engine undoubtedly holds a unique position.

PERFORMANCE OF VARIOUS TYPES

In making comparisons of performances of representative engines, careful note should be made of (*a*) the speeds and cylinder sizes of the units, and (*b*) the maximum output, expressed in pounds per square inch B.M.E.P. in each case. A comparison of performances is shown in Table I, on page 98.

TABLE I.—COMPARISON OF PERFORMANCE OF PRIME MOVERS

Type of Prime Mover	*Brake Thermal Efficiency (per cent.)*	*Overall Efficiency*		*Specific Output lb. per square inch)*
		Full Load (per cent.)	*Half-load (per cent.)*	
C.I. oil engine	37·0 (39·8)	—	—	90
Gas engine (ratio 6 to 1) . . .	26·8 (30)	22·0	17	60
Gas engine (ratio 13 to 1) . . .	36·5 (38·1)	29·0	24	85
Dual-fuel engine	36·5 (40·2)	29·0	21	80
Steam Turbine*	23·0	18·6	16	--

* 1,000 kVA. steam at 250 lb. per square inch and 700° F.

Overall efficiency includes boiler or producer efficiency.

All figures are based on lower heating value of the fuel, and those in brackets represent test-bed results, whilst others are average station results.

THE FUTURE OF THE GAS ENGINE

All piston engines are to-day overshadowed by the rapid development of the combustion turbine, but it can safely be said that it will be some years before this machine approaches even the steam turbine in efficiency. It is extremely unlikely that the overall efficiency of the combustion turbine will approach that of even the conventional gas engine for years, especially in small units (up to 1,500 B.H.P.). Below about 5,000 B.H.P. the space occupied is less than the combustion turbine, and much less than that occupied by some of the more recent proposals. Compared with the oil engine, the latter stands or falls on the availability of refined oil at a reasonable cost. Normal consumption of liquid fuel is very high, and is increasing, as are fuel-oil costs. The consumption of the world's oil has been greatly accelerated by war, and at the same time many sources have been damaged or destroyed. The modern gas engine offers an alternative prime mover, which in many cases can use "waste" gases. If present experiments on the underground distillation of low-grade coal "in the seam" are successful, a very strong case for gas-engine installations of large power at the pit head can be put forward. There is at least one gas-engine station in the U.S.A. with 200,000 B.H.P. of installed plant and several of 50,000 to 100,000 B.H.P. These use natural gas.

In other circumstances, almost any hydrocarbon can be used to produce gas. The dual-fuel engine has solved the main disadvantage of the "suction" producer, i.e. lighting up. With this type of engine, the initial start can be made on oil, and the pipes so arranged that the engine can itself produce the draught necessary to blow up the fire. As soon as gas is made in sufficient quantity and quality, the engine can be turned over without any dropping of load.

T. D. W.

GAS PRODUCERS

THE fuels that can be used in a gas producer are legion. It may be of interest to mention some of the more unusual. Gas can be and is made from the waste products of timber, such as bark, twigs, and shavings, that are found in lumber camps and sawmills; from dry grass and leaves; from all sorts of nutshells; cotton pods, tea prunings, and rice husks are frequently used in the East; whilst as a final resort, dry wine scum and animal dung have been employed. Naturally, the majority of operators prefer to use wood, charcoal, coke, or coal, but these other less-orthodox fuels may awaken the imagination to the possibilities, especially as most of these fuels are self-replacing. In many cases, of course, town's or coke-oven gas is available; the use of waste gas, such as sewage gas, blast-furnace gas, and oil-refinery waste gas has already been referred to.

Gas, as a fuel, is as cheap as, if not cheaper than, oil. For small units, the process "coal-producer-gas engine-generator bus-bars" is frequently cheaper than purchased electricity. Engine for engine, the modern gas engine is the equal of the oil engine in most respects, and it is superior to many. Modern developments have made the gas engine again a serious competitor in the field of individual prime movers, and even for large stations.

The type of gas producer chiefly in use for small and medium powers is known as the suction type, and is generally on the up-draught principle, though some types of down-draught producers are used for special purposes. Forced-draught producers, or pressure producers, are usually employed for large powers or for the production of gas for heating or process work. Occasionally suction-type producers are operated by an exhausting fan instead of the suction of an engine, the gas then being used for other than power purposes.

General Method of Operation

The operation of a producer consists in drawing or forcing through a bed of incandescent fuel a mixture of air and water vapour or steam. The steam is produced generally by the waste heat of the gas, but in some cases a small boiler or evaporator forms the top of the producer. The air supply to the fire is drawn through this evaporator and takes up steam on its way, the mixture of air and steam then being taken up through the fire.

Anthracite Fuel

In the majority of the suction-gas producers used with engines, the fuel is anthracite in the form of small beans, a trade term for pieces of coal measuring about ¾ in. by ¾ in., and having a calorific value of 13,000–14,500 B.Th.U.s per

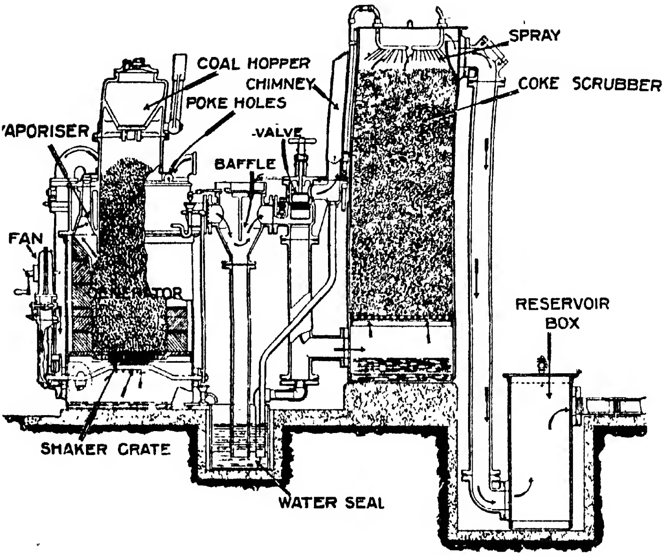


FIG. 1.—SUCTION-GAS PRODUCER (*Ruston & Hornsby, Ltd.*)

lb. The gas evolved has a calorific value of about 145 B.Th.U.s per cubic foot and a composition approximately as follows: H, 15·5; CH_4, 2·15; CO, 22·2; N, 54; CO_2, 5·1; O, 0·7. Such a gas used in a gas engine requires about an equal volume of air for combustion, and this air is added at the engine.

Suction-gas Producer

One of the simplest types of suction-gas producers is shown in Fig. 1.

The water evaporator will be seen forming the top of the producer, a flat annular chamber, the coal shoot passing through the centre of it. A drip of water is allowed to fall into a cup which communicates with the vaporiser, and an overflow pipe from the vaporiser takes any surplus water down to the ashpit under the fire.

Air enters at the elbow seen on top of the vaporiser and, passing round the vaporiser, picks up steam, and the combined air and steam pass down the pipe at the side to the ashpan. The cock in this pipe is to prevent the blast from the fan being lost by passing up to the vaporiser when the fire is being blown up, the cock being shut off during the process.

Scrubber

The gas from the producer passes out at the large branch seen on the right and down to the bottom of the scrubber. The scrubber is filled with coke, in two or more divisions. Water is sprayed over the coke from a perforated pipe

FIG. 2.—OPEN-HEARTH GAS PRODUCER

Note the two grates or trays below the producer. The automatic water-supply device is shown on top of the vaporiser. Note waste cock for standing. The suction fan is not shown. (*Crossley Bros., Ltd.*)

and runs down through the coke while the gas is passing up, and the gas is thoroughly cooled and scrubbed clean from all dust or deposits.

Testing Quality of Gas

When the fire is being blown up, a cock on the gas pipe is opened and the gas is allowed to escape into the air. A test cock is usually fitted, and the quality of the gas is tested as the blowing proceeds by lighting the gas and noting the colour of the flame.

Drawing up Fire by Suction

In some producers the fire is drawn up by a suction fan, this having the advantage of drawing steam through the fire and giving a gas of better quality for starting than when the fire is blown up.

In many types of producers the sensible heat of the gas is made use of to evaporate the water, the down pipe from the producer to the scrubber is jacketed, and water is fed to the jacket and afterwards passes into the vaporiser on top of the scrubber, a better supply of steam thus being obtained.

OPERATING NOTES

The method of working suction-gas producers is practically the same for all types, and is a very simple process.

Starting up

It will be assumed that a new plant is being started up after having been erected, or a used plant being put into service after standing for a long period.

Filling Scrubber with Coke

The first thing to be done is to prepare the scrubber. This is generally divided into two sections. The doors having been removed, broken coke is placed in the top division. The pieces should be about the size of an egg and should be well riddled to get rid of all dust before being put in the scrubber, and it is a good plan to well wash the coke, after it is riddled, with a hose, turning the heap over with a shovel so that all the coke is exposed to the water. It can be put in wet.

The lower part of the scrubber is filled with larger pieces of coke, about the size of half a brick, and these, too, should be riddled free from dust and washed.

The coke should not entirely fill the spaces; a few inches should be left from the top of the coke to the spray pipe and the perforated plate.

Making the Door Joints

The door joints should be carefully made. A sheet-rubber-insertion joint is best, as there is no heat to vulcanise it or make it stick and it will stand remaking many times. There is no internal pressure; any leakage would be of air inwards. Scrubbers are made of sheet steel, and the door-joint surfaces are often not of the best. In some cases they are not machined, so a soft material like rubber makes a better joint than sheet asbestos or one of the vulcanised asbestos rubber jointings.

Fill the Sump

When the joints are made, turn the water on to the top of the scrubber and see that the sump or trap under the vaporiser fills up, then turn the water off.

Run Water into Vaporiser

If the plant has a vaporiser on the producer, turn the water on to the drip supply and allow water to run into the vaporiser until it overflows into the cup leading down into the ashpan.

If the plant has a separate vaporiser, this should be filled; a tap is usually fitted on the last chamber to show when the water has filled to the correct level.

Lighting the Fire

The producer should be quite free of fuel or ash. If of the closed-grate type (Fig. 1), both the lower doors are opened and the slide and cover of the

fuel hopper. A handful of oily waste or some shavings are placed on the firebars and then some sticks of wood broken into fairly short lengths. The cock on the escape pipe should be opened, and the cock on the vapour pipe above the fan closed to prevent the blast from the fan entering the vaporiser instead of passing up through the fire. The fuel slide should be closed and a small charge of fuel placed in the hopper. The fire should now be lighted, and the door on the fire chamber closed, the ashpit door being kept open for a few seconds to allow the fire to draw up.

Building up the Fire

Once it has got hold, the ashpit door should be closed and the fan worked very gently. When the sound indicates that the wood is well alight the fuel slide should be drawn and a small quantity of fuel fed in. Blowing is continued and more fuel fed. Too much fuel should not be added until a good bright bed of fire is built up. The temperature of the vaporiser should be noted with the hand and blowing continued until steam is seen to emerge at the air elbow on the cover of the vaporiser, see Fig. 1. More fuel should now be added and blowing continued.

Testing Quality of Fuel

On the escape pipe just below the shut-off cock, a test cock is generally fitted (Fig. 1). This is opened and the quality of the gas is tested with a light. When the gas burns freely with an orange or mauve colour, the producer is ready for work.

Most of the troubles of difficult starting are due to trying to start the engine with the producer too cold, or the gas too poor in quality to burn. The flame from the test cock should be a solid cone of flame. If it has a long interior cone of what looks like smoke, the producer is not hot enough, or there is little or no hydrogen in the gas, a trouble to which closed-hearth producers are prone at starting. A little water should be turned on at the supply to the vaporiser and allowed to run into the ashpan from the overflow. The fan should again be operated and gas tested at the cock.

When the right quality gas is obtained, it should be impossible to blow out the flame at the test cock no matter how hard the fan is operated.

Getting ready to start the Engine

When ready to start the engine, close the test cock, also the waste cock. Adjust the water supply to the vaporiser so that a steady drip occurs at the overflow. The test cock on the gas main just below the engine gas cock should be opened, and the fan blown until gas emerges. This clears the air out of the gas main. The gas can again be tested for quality if desired. The cock on the vapour pipe above the fan is now opened, which allows the air drawn in at the elbow on the vaporiser to pass down to the ashpan, and the plant is ready for work.

D. J. S.

OPERATING NOTES

The method of working suction-gas producers is practically the same for all types, and is a very simple process.

Starting up

It will be assumed that a new plant is being started up after having been erected, or a used plant being put into service after standing for a long period.

Filling Scrubber with Coke

The first thing to be done is to prepare the scrubber. This is generally divided into two sections. The doors having been removed, broken coke is placed in the top division. The pieces should be about the size of an egg and should be well riddled to get rid of all dust before being put in the scrubber, and it is a good plan to well wash the coke, after it is riddled, with a hose, turning the heap over with a shovel so that all the coke is exposed to the water. It can be put in wet.

The lower part of the scrubber is filled with larger pieces of coke, about the size of half a brick, and these, too, should be riddled free from dust and washed.

The coke should not entirely fill the spaces; a few inches should be left from the top of the coke to the spray pipe and the perforated plate.

Making the Door Joints

The door joints should be carefully made. A sheet-rubber-insertion joint is best, as there is no heat to vulcanise it or make it stick and it will stand remaking many times. There is no internal pressure; any leakage would be of air inwards. Scrubbers are made of sheet steel, and the door-joint surfaces are often not of the best. In some cases they are not machined, so a soft material like rubber makes a better joint than sheet asbestos or one of the vulcanised asbestos rubber jointings.

Fill the Sump

When the joints are made, turn the water on to the top of the scrubber and see that the sump or trap under the vaporiser fills up, then turn the water off.

Run Water into Vaporiser

If the plant has a vaporiser on the producer, turn the water on to the drip supply and allow water to run into the vaporiser until it overflows into the cup leading down into the ashpan.

If the plant has a separate vaporiser, this should be filled; a tap is usually fitted on the last chamber to show when the water has filled to the correct level.

Lighting the Fire

The producer should be quite free of fuel or ash. If of the closed-grate type (Fig. 1), both the lower doors are opened and the slide and cover of the

Consideration must be given to the required speed, bearing in mind that the slower the speed of the motor for its power, the larger the frame size, the greater the cost, and probably the longer delivery. Speed may be fixed, and this is the least expensive type; if it is to be varied, the motor manufacturer will wish to know the speed range as well as a quantity of information detailed in a later paragraph.

Preparing to Place the Order

Of course, the installation of one or more new motors involves rather more than placing the order for the motor alone. A range of other equipment will be required before complete installation can be carried out: switch, fuses, or circuit breakers, a starter, and perhaps speed control; instruments, a switchboard or steel structure on which the foregoing equipment can be mounted—all these are included in the electrical requirements, quite apart from the cables and cable accessories to connect the motor to the main supply and sometimes an additional panel for the main switchboard.

If the general factory supply is not exactly as required by the new motor, a transformer, or a rectifier, or a motor generator may be required.

When all these points have been considered and decided, all the orders should be placed as early as possible; every effort must be made to obviate the possibility of the motor arriving ready for immediate installation, without everything else required being ready on the spot, including preparation of the foundations, cables run, and switch-gear connected up.

Specification of Requirements

The motor is the most complex of the equipment to be ordered, and may be considered first. Most firms who make motors will have a standard range of sizes, and the required h.p. and speed may fall between two of these sizes; if so, the next size above may be used, running at slightly reduced efficiency, and available of course for increased load should that become necessary at a later date; or alternatively the size below should be chosen on the assumption that the required load may be intermittent or may not always reach motor full load; or again, whether a specified overload may be permitted for a specified time.

Information Required by Motor Makers

Almost any list showing the range of electric motors available will provide a schedule of the information required by the makers of the motor before they can quote. Some of it may already be known to them; for example, they are bound to be aware of the local supply voltage, and whether it is A.C. or D.C., of standard frequency or not, and whether single-phase or three-phase.

It might be thought that the h.p. comes next; but for a given h.p. the "frame size" may vary according to continuous or intermittent running, a load factor if the load is varying, and speed variation if necessary; all these and even the power factor may have to be considered before a decision on the frame

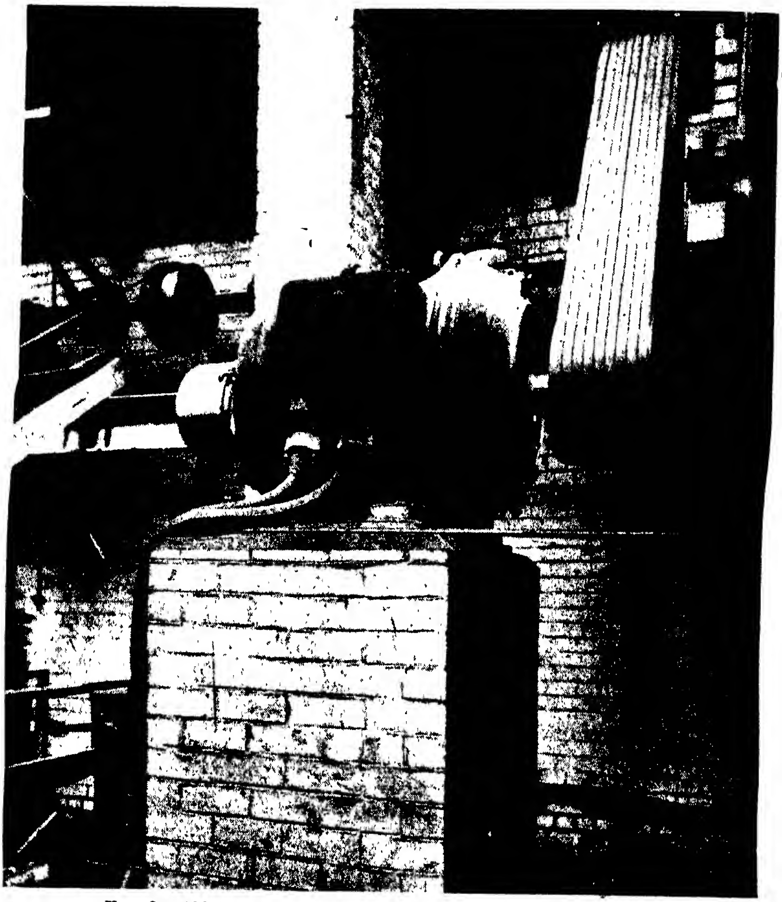


FIG. 2.—100-H.P. MOTOR INSTALLED ON BRICK AND CONCRETE PIER
The motor is driving a line shaft in a brick works through a rubber V-belt.
(*The British Thomson-Houston Co., Ltd.*)

size can be made. If there is any possibility of an overload, this also should be stated.

The higher the speed, the smaller the frame; it might even pay to run at a high speed and reduce the required speed on the output shaft by means of gearing, or geared down-drive by belt, ropes, or chain. Variation in speed required should invariably be stated, as this affects the type of motor most suitable for the work; and if there is no variation required, this also should be stated, as a less expensive motor will be suitable.

The makers of the motors will require full information in regard to the purpose to which the motor is to be applied and the general conditions under which it will operate. The leading makers of electric motors issue leaflets in which they ask an enquirer to state the atmospheric conditions, and to supply information in regard to variation of voltage and possible voltage drop due to long cable runs, together with information on starting, load, type of drive, and any other speed requirements.

Atmospheric Conditions

Dealing first with the atmospheric and other surrounding conditions, a motor for use in a mine, gas works, or chemical industry, petrol stations or even a flour mill may have to be of the totally enclosed type, and possibly flameproof as well; motors for damp situations, or out of doors, would have to be weatherproof, or dripproof, and even in a dry atmosphere, such as that of a flour mill or chemical works, a dustproof enclosure may be necessary. The standard range of enclosures is open type, protected (as by an expanded metal screen), dripproof, totally enclosed, and flameproof.

The latter types may require special air cooling in order to keep the temperature down despite the total enclosure; where there is no risk of gas or dust explosion, a fan may be mounted on the rotor shaft, and this type of motor is styled "fan cooled." Where the motor is to be flameproof, and have no connection with the outer air, cooling may be by means of a flow of ventilating air brought to the motor from a safe source of supply by means of a pipe, and the motor is then styled "pipe cooled."

The maker would also like to know, for any motor which is not running continuously throughout an eight-hour day, the frequency of starting, how often reversing is necessary, and whether there are special conditions at starting requiring a long running up period, a high starting current, or powerful torque.

There remain only those mechanical conditions affecting the frame rather than the electrical part of the motor. Most motors have a horizontal shaft, and the maker will want to know the direction of rotation required, and whether the shaft is to be extended at each end; for certain special conditions, a motor with a vertical shaft may be the most suitable, driving direct or through a gear.

General Arrangement and Installation Plans

It has been assumed that all the orders will have been placed on the basis of the longest delivery, which might be the motor, or perhaps the electronic control gear, where this is used. Long before delivery takes place with anything but small stock-size motors, the maker of the equipment will be able to supply general arrangement blue prints and installation plans. These will indicate, not only the floor space to be taken up by the main items, but the suggested relationship of starters and control panels to the position in which the operator stands or sits.

The installation drawing will provide bedplate or other fixing details; but it will be for the mechanical engineer to ensure that when the equipment is

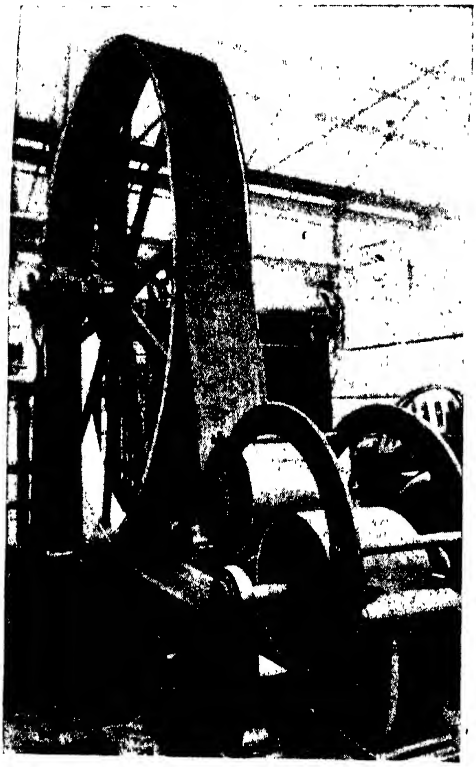


FIG. 3.—LARGE-RATIO DRIVE AT SHORT CENTRES WITH JOCKEY PULLEY

The jockey pulley tightens the belt round the pulley and increases the arc of contact. The drive is capable of 150 h.p. and a belt speed of 5,100 ft. per minute. The balata 6-ply belt is 16 in. wide.

fixed in position, perhaps close to other equipment or machine tools, it is still possible for crane or hoist to lift out any parts of the electrical equipment. That is, the highest point of the equipment should not be so close to any roof structure that block and tackle cannot be arranged to lift off for example the top half of the stator, or if necessary the whole motor.

On larger machines it may be necessary in emergency to lift out the rotor; and this involves, not only the necessary head room for lifting equipment, but if the stator is in one piece, the rotor will require room at the end to be moved out before hoisting begins.

The general arrangement and installation drawings will also show how the drive of the motor is to be applied to its work; it may be by the ordinary pulley to line-shaft, and if the pulley is a large one, or the shaft extension is considerable, a third bearing styled a "pedestal" bearing may be necessary; or the motor may be direct-coupled to its load.

Preparation for Installation

Once the site has been decided upon, and a local layout plan prepared to show the relation of the motor and machine to the switches, starter, and other control equipment, some further preparation can be carried out which will save time after the motor has been delivered.

For example, if the starter and switchgear, less expensive equipment than the motor and therefore of shorter delivery, have been delivered, they can be installed; the conduit can be coupled up as far as the switchgear and starter,

and insulation and continuity tests made to ensure that the cable runs up to that point are satisfactory.

The conduit may even be run up to the point at which it will emerge from the concrete floor for connection to the motor, provided that the exact details of the motor terminal box are known; if there is any doubt, this part of the work should not be finished off until the motor arrives.

After the Motor is Received

When the motor is received and unwrapped, it is thoroughly wiped down and cleaned; and a careful examination must be made to ensure that there has been no damage in transit (which should be notified to the makers) and that no parts have been loosened or detached during transit: this examination should include a check that the name-plate corresponds to the order. The shipping notices should be carefully retained; makers like to know the number of the workshop order to which the machine has been built in case of any query as regards condition, or for use when ordering spare parts. If this is mislaid, they will still be able to identify the motor from the serial number stamped on the nameplate.

Testing Insulation Resistances

The insulation test should then be made; this will show whether drying out in a warm place is necessary. The insulation resistances in megohms measured between any terminal and the frame, with the machine cool, should not be less than:

$$\frac{\text{rated voltage}}{1000 + \text{rated B.H.P.}}$$

The time to dry out may take one or several days, and in either case the warming should be continuous, with frequent and regular tests of insulation resistances and temperature. The temperature should not exceed 80° C., and should not be allowed to fall unduly, in case moisture should be reabsorbed.

Redistribution of moisture causes the insulation resistance to fall until a minimum value is reached; it may then remain constant for a period of hours or days before beginning to rise to its maximum and final value.

Methods of Drying Out

The simplest method of drying out is to place the machine in a situation where the atmosphere is dry, free from dust, and warm, with a temperature not exceeding 80° C. (176° F.). This method may be improved by boxing in the motor and circulating hot air through it. The air must be thoroughly dried before heating by passing it through a filter or drying chamber of dry, clean lime, or calcium chloride, and should be distributed by means of baffles to ensure equal distribution and equal temperature inside the box.

In many cases it will be difficult to find such a situation, and in these instances carbon lamps or heaters may be placed both around and inside the motor. Great care should be taken to see that winding temperatures in the vicinity of the heaters do not attain a dangerously high level.

Where skilled personnel and appropriate means are available, the usual and best way to dry out a motor is to block the rotor so that it cannot turn round, and apply a very low voltage, about 10 per cent. of normal, to the stator terminals. In the case of slipring motors, the rotor windings must be shortcircuited. In this case the heating current must be switched off from time to time to enable the insulation resistance to be measured, but if the machine is not thoroughly dry, it should be switched on again immediately.

Close supervision is necessary during the process of drying out with electric current. The heat generated in the windings is not easily dissipated, and one part of the winding may be exceedingly hot before another part has had time to expel the moisture. This can be obviated to some extent by taking every precaution to exclude draughts from the exposed parts of the windings. It may even be necessary to cover such parts of the windings with wrappings.

Lubrication of Bearings

With ball or roller bearings, it is certain that complete lubrication will have been provided by the makers; instructions given with the machine will state the type of grease or oil to be used for replenishment. If the bearings are of sleeve type, these will require to be filled up with the oil recommended in the maker's instruction. When this is being done, it is usual to turn the rotor by hand, if possible, to ensure that the oil rings are revolving freely and actually bringing the oil up.

Storing of Motor

The motor is now ready for installation; but supposing that it is not to be used almost immediately, it may be desirable to store it in a warm room where it could not possibly absorb moisture again. This will obviate damage or dust which may result from the motor being erected and left installed in a building which may either be under construction or in process of alteration.

Foundation Work

When mounting at floor level, the machine and bedplate, or slide rails, should be securely bolted on a solid and level foundation. Concrete or solid masonry is the most desirable bed, and all foundation plates and rag bolts should be securely grouted in.

When making a concrete foundation the excavation should be carried well down. The depth of foundation depends upon the nature of the ground, but assuming average solid ground which has been well rammed, both below and around the excavation, the following depths are figures for average drives. Generally, motors above 100 h.p. require to be considered with their individual drives:

10–25 h.p. . . .	6–8 in. deep.
25–50 h.p. . . .	8–10 in. deep.
50–75 h.p. . . .	10–15 in. deep.
75–100 h.p. . . .	15–24 in. deep.

Round the edges of the excavation and at floor level a wooden mould should be erected to shape the plinth. Any cable ductings or pipelines should also be provided for at this stage by the insertion of suitable wooden moulds. Attention should be paid to the fixing of the foundation bolts: they may either be fixed in position and the concrete cast round them or holes may be left in the concrete base and the bolts afterwards grouted in. The latter is the more common method.

A good foundation cement consists of 1 part cement, 2 parts sharp sand, 4 parts broken stone.

The parts should be thoroughly mixed whilst dry, water then being slowly added through a "rose" until the stones are well coated with the cement and sand. The mould should be filled with the mixture and well rammed down. When the concrete has set, the wooden mouldings should be removed, and if necessary any small defects and holes patched up.

Levelling and Grouting of Bedplates

It is important that all bedplates, whether for individual motor units or for complete sets, should be properly levelled by means of steel wedges before being grouted in.

In the case of individual units, the machine and its bedplate or side rails should be placed in position on the foundations, and lined up with reasonable accuracy with the driven or driving shaft. The baseplate or slide rails should be packed up between the bed and the concrete or masonry by steel wedges placed at intervals of 12 in or less along the bed, and the holding-down bolts, provision for which should have already been made, grouted in position.

When the grouting mixture round these bolts has set hard, the baseplate with the motor in position should be levelled up and, if found to have warped since being machined, should be sprung level by means of the wedges. The foundation bolts should then be tightened down and the alignment again checked. Shaft alignment is dealt with later.

Heavy Installations

For heavy installations, such as large motor generator sets which are mounted on complete bedplates, it is preferable to have a number of steel girders sunk into the foundations in order to reinforce the concrete, the number depending on the size and weight of the set. The uppermost flange should be machined and the girders laid across the base, projecting slightly above the level of the concrete. Some of the girders should be positioned so as to lie under all the heavy-load points of the bedplates.

A levelling process similar to that previously described should be adopted for these sets, with steel wedges placed at intervals of 12 in. or less along the bed between the bedplate and the concrete, and between the bedplate and the girders.

After the concrete foundations have been laid, it is advisable to leave the set for a period of days to allow for settling before the final checking and grouting in of the bedplate.

Where skilled personnel and appropriate means are available, the usual and best way to dry out a motor is to block the rotor so that it cannot turn round, and apply a very low voltage, about 10 per cent. of normal, to the stator terminals. In the case of slipring motors, the rotor windings must be shortcircuited. In this case the heating current must be switched off from time to time to enable the insulation resistance to be measured, but if the machine is not thoroughly dry, it should be switched on again immediately.

Close supervision is necessary during the process of drying out with electric current. The heat generated in the windings is not easily dissipated, and one part of the winding may be exceedingly hot before another part has had time to expel the moisture. This can be obviated to some extent by taking every precaution to exclude draughts from the exposed parts of the windings. It may even be necessary to cover such parts of the windings with wrappings.

Lubrication of Bearings

With ball or roller bearings, it is certain that complete lubrication will have been provided by the makers; instructions given with the machine will state the type of grease or oil to be used for replenishment. If the bearings are of sleeve type, these will require to be filled up with the oil recommended in the maker's instruction. When this is being done, it is usual to turn the rotor by hand, if possible, to ensure that the oil rings are revolving freely and actually bringing the oil up.

Storing of Motor

The motor is now ready for installation; but supposing that it is not to be used almost immediately, it may be desirable to store it in a warm room where it could not possibly absorb moisture again. This will obviate damage or dust which may result from the motor being erected and left installed in a building which may either be under construction or in process of alteration.

Foundation Work

When mounting at floor level, the machine and bedplate, or slide rails, should be securely bolted on a solid and level foundation. Concrete or solid masonry is the most desirable bed, and all foundation plates and rag bolts should be securely grouted in.

When making a concrete foundation the excavation should be carried well down. The depth of foundation depends upon the nature of the ground, but assuming average solid ground which has been well rammed, both below and around the excavation, the following depths are figures for average drives. Generally, motors above 100 h.p. require to be considered with their individual drives:

10–25 h.p.	6–8 in. deep.
25–50 h.p.	8–10 in. deep.
50–75 h.p.	10–15 in. deep.
75–100 h.p.	15–24 in. deep.

Belt Drives

When a belt is used, the joint in the belt should be smooth and flexible and the pulley should be well balanced. Vertical belt drives are to be avoided, and belt drives with very short centre distances cannot be considered good practice unless some form of idler or jockey pulley is employed to increase the arc of contact on the driving pulley, and to keep the belt tension at the correct value (see Figs. 3 and 5). Generally, in drives without idler or jockey pulleys, the pulley centres should not be less than four times the diameter of the larger pulley.

Tables I and II give the horsepower transmitted by single leather and balata belting respectively.

For light double leather belts the figures given in Table I may be increased by 50 per cent. and for heavy double leather belts by 80 per cent.

Belts should be chosen to carry 100 per cent. above normal motor rating for direct-started motors and 25 per cent. above for all other types of motors.

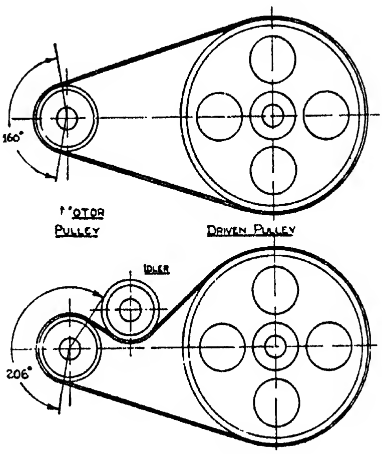


FIG. 5.—THE USE OF AN IDLER PULLEY

In short centre distance or large ratio belt drives, the use of an idler pulley will, very often, make an otherwise bad and troublesome drive perform satisfactorily Observe how the arc of contact on the motor pulley has been increased by nearly 30 per cent.

Measuring Belt Length

In fitting a belt, the length may be measured with a string or steel tape, or the following formula will serve to give approximate results where open belts are involved:

Belt length $= 3{\cdot}14\dfrac{(D+d)}{2}$ + twice the distance between shaft centres, where—

D = Diameter of larger pulley.
d = Diameter of smaller pulley.

The belt should be sufficiently tight to carry the load without flapping on the slack side, as this is dangerous and also results in a noisy drive. On the other hand, excessive belt tension invites bearing trouble or shaft failure, and is responsible for uneven running and a short belt life. Sliding bases should be employed when a belt drive is used.

Belt Speed

It is impossible to give hard-and-fast rules as to the correct belt speed, as this is entirely dependent upon the load. A good average is 4,000 ft. per minute; but this may be as low as 3,000 ft. per minute on a stone-crushing load and as high as 7,000 ft. per minute with a fan.

TABLE I.—HORSE-POWER TRANSMITTED BY SINGLE LEATHER BELTING

Velocity (ft. per minute)	*Width of Belt (in.)* 2	4	6	8	10	12
200	0·60	1·20	1·80	2·40	3·00	3·60
300	0·90	1·80	2·70	3·60	4·50	5·40
400	1·22	2·44	3·66	4·88	6·10	7·32
500	1·52	3·04	4·56	6·08	7·60	9·12
600	1·82	3·64	5·46	7·28	9·10	10·92
700	2·12	2·24	6·36	8·48	10·60	12·72
800	2·42	4·84	7·26	9·68	12·10	14·52
900	2·72	5·44	8·16	10·88	13·60	16·32
1,000	3·02	6·04	9·06	12·08	15·10	18·12
1,500	4·54	9·08	13·62	18·16	22·70	27·24
2,000	6·06	12·12	18·18	24·24	30·30	36·36
2,500	7·58	15·16	22·74	30·32	37·90	45·48
3,000	9·00	18·00	27·00	36·00	45·00	54·00
3,500	9·80	19·60	29·40	39·20	49·00	58·80
4,000	10·18	20·36	30·54	40·72	50·90	61·08
4,500	10·14	20·28	30·42	40·56	50·70	60·84
5,000	9·65	19·28	28·92	38·56	48·20	57·84
5,500	8·95	17·90	26·85	35·80	44·75	53·70

(*Metropolitan-Vickers Electrical Co., Ltd.*)

TABLE II.—HORSE-POWER TRANSMITTED PER INCH WIDTH OF BALATA BELTING

Velocity (ft. per minute)	3-*ply*	4-*ply*	5-*ply*	6-*ply*	7-*ply*	8-*ply*	9-*ply*	10-*ply*
500	0·62	0·83	1·04	1·26	1·48	1·65	1·81	1·97
1,000	1·25	1·66	2·09	2·50	2·92	3·33	3·73	4·13
1,500	1·87	2·50	3·12	3·75	4·37	5·00	5·65	6·30
2,000	2·50	3·33	4·16	5·00	5·83	6·66	7·45	8·25
2,500	3·12	4·16	5·20	6·25	7·29	8·33	9·35	10·37
3,000	3·75	5·00	6·25	7·50	8·75	10·00	11·25	12·50
3,500	4·37	5·83	7·28	8·75	10·20	11·66	13·03	14·50
4,000	5·00	6·66	8·33	10·00	11·66	13·33	14·97	16·62
4,500	5·62	7·50	9·37	11·25	13·12	15·00	16·87	18·75
5,000	6·25	8·36	10·31	12·51	14·58	16·66	18·77	20·89
5,500	6·88	9·17	11·46	13·76	16·64	18·34	20·64	22·92

(*Metropolitan-Vickers Electrical Co., Ltd.*)

TABLE III.—RELATION OF LEATHER BELT THICKNESS TO PULLEY DIAMETER

Smallest Pulley Diameter .	*Up to* 6 *in.*	6–14 *in.*	14–22 *in.*
Type of belt . . .	*Single*	*Light double*	*Heavy double*

(*Metropolitan-Vickers Electrical Co., Ltd.*)

TABLE IV.—RELATION OF BALATA BELT THICKNESS TO PULLEY DIAMETER

Belt ply . .	3	4	5	6	7	8	9	10
Pulley diameter (in.)	3	4	12	16	18	24	30	36

(*Metropolitan-Vickers Electrical Co., Ltd.*)

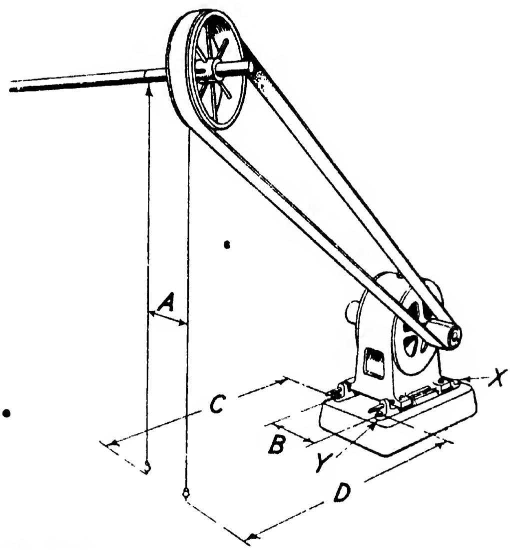


FIG. 6.—ALIGNING A BELT DRIVE

For true alignment between the shafts the distances *C* and *D* must be equal. Note that the measurement of *C* or *D* must be made on a line at right angles to the motor shaft. The easiest way to ensure this is to measure along a straightedge which is laid across the centres of the foundation bolts of each slide-rail, i.e. bolts *X* and *Y*. In other words, the plumb bob, bolt *X*, and bolt *Y* must lie on a straight line. Distances *A* and *B* should of course be equal, and this can be ensured by using the lining-up method illustrated in Fig. 7. A steel rule or tape should be used in taking measurements.

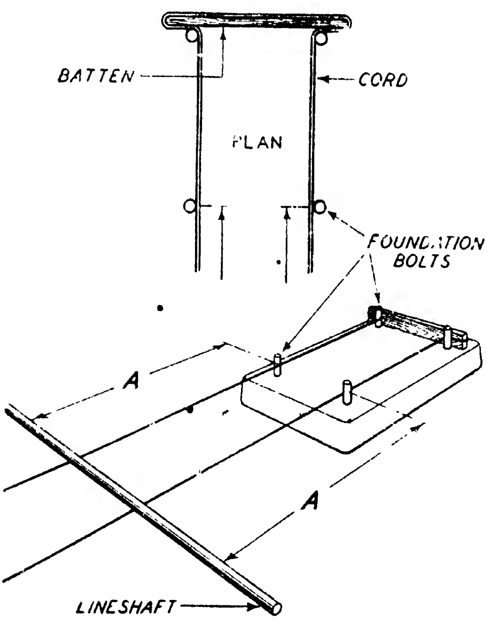


FIG. 7.—USING A LINING-UP CORD FOR ALIGNING BELT, ROPE, CHAIN, OR GEAR DRIVE

This shows a method of ensuring that measurement *A* is taken along line at right angles to the motor centre line. If the measured distances *A* are the same, the shafts are in line. The cord (or wire) is stretched tightly between the motor foundations and supports some distance beyond the line-shaft. The sketch shows a very convenient method of using the cord.

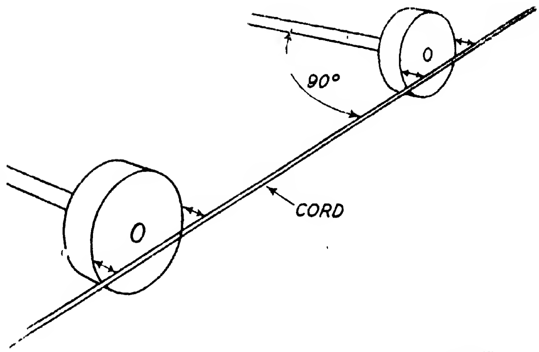


FIG. 8.—ALIGNING PULLEYS—CORD METHOD

The cord is stretched at right angles to one of the shafts and in a position a few inches from the edge of the pulley. The position of the second shaft and of the pulley upon it is then adjusted, so that the four dimensions shown arrowed are equal, and the pulleys are then truly aligned.

With belt drives generally it is very necessary to ensure that the driving and driven shafts are parallel, and the driving side of a fast and loose pulley arrangement should always be next to the machine bearing.

Shaft Alignment

For correct alignment the two shafts should be parallel and the centres of the pulleys directly opposite. To check whether the shafts are parallel, the shortest distances between the driving and driven shafts should be measured in two places along their axes, as far apart as the smaller of the shafts will allow; the measurement should preferably be made with a steel tape or some other form of inextensible measure.

Pulley Alignment

The usual way to ensure that the pulleys are directly opposite is by the use of a wood or steel straightedge (Fig. 9). If, as in the majority of cases, both pulleys have the same width of face, then the straightedge should be laid across the edge of the rim of the driven pulley, touching it in two places. The driving pulley should then be adjusted until, with the straightedge lined up against the driven pulley, the edge of the rim of the second pulley also touches in

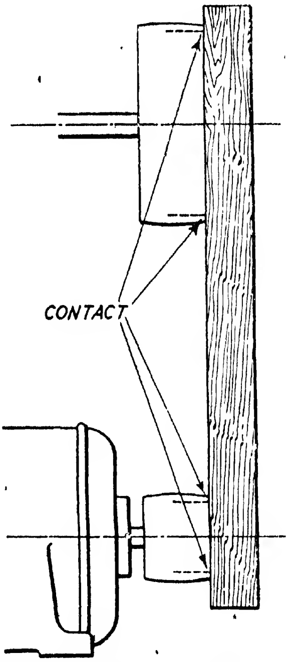


FIG. 9.—ALIGNING PULLEYS—STRAIGHTEDGE METHOD

The straightedge is placed so as to make contact with both pulleys at two points at opposite ends of diameter in each case. If pulley faces are unequal, the cord method (Fig. 8) should be used and centre of pulleys aligned.

two places. In those cases where the distance between shaft centres is too great to employ a straightedge, the cord method should be used (Fig. 8).

When the set is lined up and the two shafts are directly opposite to each other, the machine should be started up to see that the belt runs freely. If the shafts are parallel but the pulleys not directly opposite, the belt will run to one side of the larger pulley. If the pulleys are opposite, but the shafts are not parallel, the belt will run to one side of the smaller pulley.

Rope Drives

The general remarks quoted under belt drives also apply to rope drives. The maximum horse-powers transmitted by rope drives at various speeds are given in Table V. A conservative allowance has been made for the reduction of power due to centrifugal force at the higher speeds, although in practice such a reduction is not always considered necessary.

Table VI indicates the size of rope suitable for drives where the diameter of the smallest pulley is not less than the figure given in the table, and the peripheral speed is approximately 5,000 ft. per minute.

Gear and Chain Drives

When the motor is to drive by means of gears or chains, it is essential that the alignment be exact. All wheels should be fully meshed both in depth and along their width of face, and chains must run centrally on their sprockets. For gear drives with pitch-line speeds exceeding 1,000 ft. per minute, paper or fabric pinions are recommended to avoid excessive noise. Faulty alignment and bad meshing of gear and chain drives may give rise to destructive vibration, or bearing trouble, together with a bent or broken shaft. Machines built on the

TABLE V.—HORSE POWER TRANSMITTED BY ROPES AT VARIOUS SPEEDS

Velocity (ft. per minute)	*Diameter of Rope (in.)*						
	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
1,000	2·5	3·5	4·5	7·1	10·2	14·0	18·2
1,500	3·7	5·4	6·7	10·6	15·3	21·0	27·1
2,000	5·1	7·0	9·1	14·2	20·5	28·0	36·5
2,500	6·2	8·4	11·0	17·2	24·7	33·7	44·0
3,000	7·1	[illegible]	12·6	19·8	28·4	38·7	50·4
3,500	7·8	10·7	13·9	21·8	31·4	42·8	55·8
4,000	8·4	11·4	14·8	23·2	33·4	45·5	59·4
4,500	8·6	11·7	15·3	24·0	34·5	47·0	61·4
5,000	8·7	11·8	15·4	24·1	34·6	47·1	61·5
5,500	8·4	11·4	14·8	23·2	33·4	45·5	59·4

(*Metropolitan-Vickers Electrical Co., Ltd.*)

TABLE VI.—RELATION OF ROPE DIAMETER TO PULLEY DIAMETER

Rope diameter (in.) . .	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
Pulley diameter (in.) . .	24	27	30	39	45	54	66

(*Metropolitan-Vickers Electrical Co., Ltd.*)

larger frames are not designed for gear or chain drives unless outboard bearing arrangements are used.

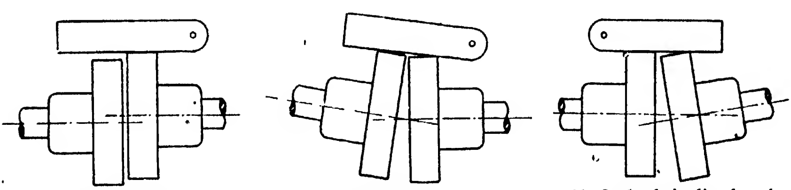
Bevel and single helical gear drives give rise to thrust which, in certain cases, may be excessive and beyond the capacity of the motor bearing. For this reason, drives of the above types should never be used without previous reference to the makers.

Direct-coupled Drives

Either solid or flexible couplings may be used when a motor is direct-coupled to its load. The flexible coupling is to be preferred, as it protects the machine from any end thrust and reduces any stress on the shaft and bearings that may arise from slight misalignment.

On all direct-coupled drives, whether flexible or solid couplings are used, correct alignment is of paramount importance. Bad alignment may quickly ruin a flexible coupling, and will undoubtedly lead to a bent shaft and damaged bearings where a solid coupling is used.

The alignment of the shafts should be checked by measuring the height of centres. This can easily be done by placing a straightedge, such as a steel rule, across the outside machined surfaces of the coupling flanges. If the centre heights are the same, the straightedge will touch both flanges along their entire lengths (see Fig. 11). The horizontal alignment may be checked in a similar manner by placing the rule across the sides of the coupling.



Shafts are parallel but out of centre.

Couplings are on same centre but shafts are inclined to each other.

Shafts both inclined and out of centre.

FIG. 10.—LINING UP COUPLINGS

The two half-couplings are here shown in various states of mal-alignment, and out of centre, and a straightedge is shown across the flanges in each case. The lack of alignment is at once obvious when the contact between straightedge and *both* coupling flanges is inspected. Note also how the distance between the two coupling faces differs at the top and bottom edges in those cases where the shafts are inclined to each other.

FIG. 11.—ALIGNING COUPLINGS (1)
The alignment of the shafts should be checked by measuring the height of centres. A straightedge placed across the coupling flanges makes full contact only when the shafts are in line, always providing that the flanges are of exactly the same diameter. (*Metropolitan-Vickers Electrical Co., Ltd.*)

FIG. 12.—ALIGNING COUPLINGS (2)
The distance between the coupling faces should be measured by a feeler gauge. At least four points should be measured twice. For the second measurement, half the coupling only should be revolved half a revolution. If all measurements are equal, the shafts are in line. (*Metropolitan-Vickers Electrical Co., Ltd.*)

The alignment should also be checked by measuring the distance between the two coupling faces with a feeler gauge (see Fig. 12). The coupling should not be turned during this measurement. The distance between the two halves should be the same all the way round. At least four points should be measured twice. For the second measurement, half the coupling only should be revolved half a revolution.

Pipe-ventilated Machines

Since the machine relies entirely on the cooling air for its successful operation, it is necessary to ensure that the pipe connection is not less than 1·5 times the area of the inlet opening to the machine, as straight as possible, with a maximum of two right-angle bends and a length not exceeding 50 ft.

Pedestal-bearing Machines

With a pedestal-bearing machine, it is necessary first to set the bedplate on its foundation; and the height is adjusted so that approximately half the total number of liners allowed for aligning the machine on its centre line can be inserted between the bedplate and the frame of the machine.

The lower half of each bearing is then placed in position, and the bearings and oil reservoirs inspected carefully to ensure they are clean and free from dirt.

The correct position of the bearings is marked for easy location during erection by means of lines, and stamped numbers are used to ensure that each pedestal shall go in its correct position. The rotor should then be threaded into the stator, taking great care not to damage the windings and making sure that the stator terminals are in the correct relative position to the sliprings.

The protective coating on the rotor shaft is removed by means of turpentine, petrol, or methylated spirit, and the journals wiped clean. The rotor and stator are lifted together, and lowered gently into position; it is necessary to ensure that the journals drop into the bearings and that the oil rings are in place. The top halves of bearings and caps are then put on.

The cap bolts are then screwed down lightly, and an inspection made to ensure that the oil rings are free to move and that the stator is not fouling the rotor. The cap bolts may be tightened after the rotor has been turned by hand, and the operation of the bearings has been found satisfactory. As already indicated, alignment of motor shaft with driven shaft must be accurately carried out before the foundations are set.

Connecting Up, and First Run

When the wiring has been completed, and tested for insulation and continuity, steps should be taken to ensure that all switches are in the off position, and all starter and control handles similarly safe. All overload releases should be set to trip at about 50 per cent. more than the full-load current stamped on the motor nameplate. Dashpots for overload trips should be filled to the correct level with the oil recommended by the makers.

The first important point is to ensure that the direction of rotation is correct; on small motors the terminals are painted in three colours, and the direction of rotation stated, providing connections are made accordingly. If the rotation is wrong, two leads must be interchanged. If the motor does not start when arriving at the second or third notch of the starter, the main switch must be opened, and then the starter handle returned to the off position for a further check-up.

On a first run special care should be taken that the temperature rise is not excessive, especially where the total load on the motor is not exactly known. Now that clip-on ammeters are available, readings can be taken without the trouble of connecting in testing instruments; and if there is any fear that the load is too high, temperature readings should be taken by means of a thermometer.

The temperature rise permissible will invariably be stated in the instruction book which goes with the machine; motors are generally constructed to operate with a temperature rise not exceeding 40° C. on cores and windings, measured above atmospheric temperature not exceeding 35° C., allowing a total temperature of 75° C. This rating permits reasonable overloads.

We wish to thank the Metropolitan-Vickers Electrical Co., Ltd., for supplying some of the material for this article.

G. W. W.

PUMPS AND PUMPING

PUMPS for lifting or conveying liquids can be broadly divided into a number of different categories, according to some distinctive feature or with reference to a particular duty, but no useful purpose is served by attempting an exact classification. Without taking into consideration special applications, the majority of pumps would come under one of the following headings:

Types of Pumps

(1) Reciprocating pumps, in which a plunger or piston is mechanically reciprocated in a liquid cylinder.

(2) Centrifugal pumps, in which pressure energy is given to the liquid by the rotation of an impeller and centrifugal force.

(3) Rotary pumps, where the liquid is forced through the pump cylinder or casing by means of rotating drums or pistons without the aid of centrifugal force.

(4) Air-lift pumps, which raise water by the buoyancy of an aerated column of water in a submerged tube.

The engineer confronted with a pumping problem obviously wishes to know the most suitable and efficient type and size of pump for that particular service. In some cases it might be possible to say that experience has shown that a particular type had given satisfactory service over a number of years, but more frequently two or more different types might be equally efficient and reliable. In order to give a satisfactory answer to such a problem, it is necessary to take into consideration all the prevailing circumstances, together with the guaranteed, or probable, efficiencies of suitable types of pumps.

Modern pumps of all types have undoubtedly reached a high degree of efficiency, but probably the greatest development has been in the range of operations performed by the centrifugal pump. For a great number of purposes the direct-coupled and the belt-driven centrifugal pump has displaced the older reciprocating pump, in spite of the fact that the efficiency of the centrifugal pump is lower. It has, however, many advantages, of which low initial cost, absence of valves, compactness, lighter foundations, and low running and repair costs are the most important. At the same time the centrifugal pump is unable to compete with the reciprocating and rotary types working against high pressure with fluctuating outputs, particularly when the heat in the exhaust steam from direct-acting or steam-driven pumps can be usefully employed.

The range of duties performed by pumps is so large that even to enumerate the trades and industries in which they are used would take up a great deal of

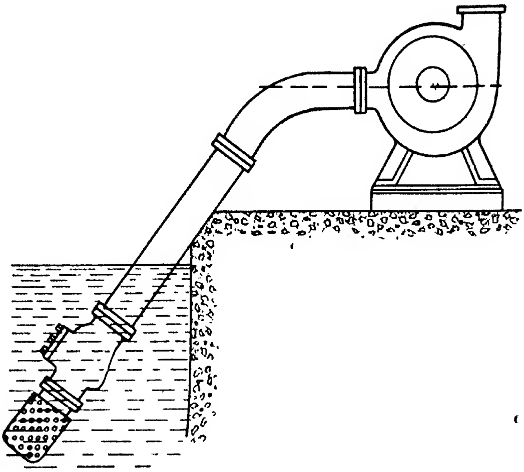


FIG. 1.—ARRANGEMENT OF SUCTION PIPE WITH FOOT VALVE AND STRAINER

space. Many pump manufacturers have designed and developed pumps for specific duties, and no doubt the specialist is best able to satisfy the demands of those requiring a special-duty pump.

The various types of pumps used in industry differ considerably in detail, and it is therefore proposed to select typical examples of modern pumping practice, as applied to water pumps, and to indicate the special characteristics and modifications.

ARRANGEMENT OF PIPES AND FITTINGS

It is of little use installing a highly efficient pump if the power to drive the pump is dissipated, or much greater than it would be, owing to the use of unsuitable pipes and fittings, or a poor layout. The overall efficiency of a pumping plant, in many cases, depends more upon the piping arrangements than upon the pump, and quite as much attention should be given to the selection of the correct pipes and fittings as to the choice of a particular type and size of pump.

Suction and Discharge Pipes

Both suction and discharge pipes should be of ample size; they should be as short and direct as possible, with few bends and no elbows or Ts, and quite free from possible air pockets. The suction pipe should rise gradually towards the pump, and should receive sufficient support to prevent vibration and take all weight off the pump inlet flange.

The arrangement shown in Fig. 1 is a good one, and makes use of a bell-mouthed entry foot valve and a strainer. The position of flanges, foot valves, and other fittings should be as accessible as possible for examination and cleaning. An alternative arrangement, and the one most frequently adopted, is shown in Fig. 2. If possible, the vertical pipe should be supported on a brick or concrete pillow.

Faults in Suction Pipes

A common fault, and one causing a good deal of trouble, is shown in Fig. 3. Here the suction pipe connected to the pump slopes upward away from

the pump, with the result that an air pocket is formed, which gives rise to loss of suction, irregular discharge, and water hammer.

A similar fault is found when reducing pipes are used to connect an oversize pipe to the pump inlet. The type of reducing pipe shown at *A* in Fig. 4 can often be usefully employed, but when used in conjunction with a 90° bend, it is difficult to obtain sufficient downward slope to prevent a bad air pocket forming in horizontal pipes. Such a pipe is quite satisfactory on the discharge side of the pump when used to increase the diameter of the piping, but for suction purposes the type of reducer shown at *B*, Fig. 4, should be used.

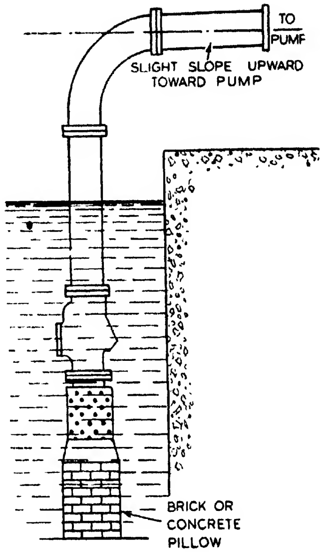


FIG. 2.—ALTERNATIVE ARRANGEMENT OF SUCTION PIPE

Foot Valves and Strainers

A foot valve is advisable when the suction pipe is of large diameter or more than about 10 ft. in length. This valve is intended to retain water in the suction pipe and enable the pump to start discharging water without first exhausting air from the pipe, or priming the pump.

When necessary, a strainer should be fitted to the inlet of the suction pipe. The area of the holes or slots should be not less than three times the sectional area of the pipe. The upper row of holes or slots require to be at least two pipe diameters below water level. When a foot valve is fitted, it is advisable to provide a drainpipe for emptying the pipe in cases where there is a possibility of freezing.

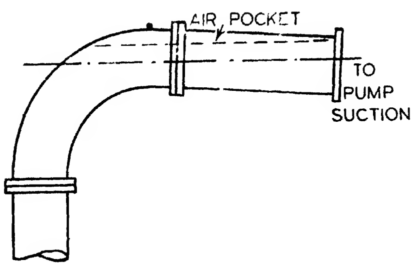


FIG. 3.—INCORRECT PIPE ARRANGEMENT—PIPE SLOPING DOWN TO PUMP

Inlet Pipe

When a strainer is not provided, the inlet pipe should be fitted with a bell-mouth entry. This allows the free entry of water and ensures a full pipe without throttling the flow of water.

Priming Pipe

This usually consists of a pipe and screw-down valve connecting the discharge pipe to the

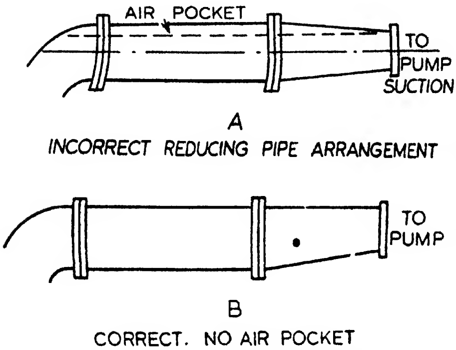


FIG. 4.—CORRECT AND INCORRECT REDUCING PIPE ARRANGEMENTS

suction pipe, by means of which water can be run into the suction pipe and pump just before starting up. Any leakage past the foot valve is made up and the system charged. Most pumps are provided with a filling plug for use when a priming pipe is not fitted.

Non-return Valve

A non-return valve is usually fitted above the discharge valve when the head against the pump exceeds about 200 ft. or when the delivery pipe is 200 ft. in length or more. This valve retains water in the delivery pipe when the pump is stopped and enables the pump to be primed by exhausting the air in the pump and suction pipe. It also prevents the pump from being damaged by water hammer caused by suddenly stopping the pump. A type of valve frequently used is shown in Fig. 5.

Clack Valve

The valve shown in Fig. 6 is fitted to the discharge of outdoor pipes when they are liable to become silted up if the pump is stopped for any length of time. The counter-balance weight allows the valve to be opened or closed as desired.

Suction Exhausters

When the suction pipe of a centrifugal pump is of large capacity and a foot valve is impracticable, the pump and piping may have to be exhausted of air by means of a plunger-pump exhauster driven by a belt or other suitable means, or by using an air pump or steam ejector. A small plunger-pump exhauster is illustrated in Fig. 7. It is essential that the delivery be closed to atmosphere by means of a sluice valve or non-return valve.

Air Vessels

Reciprocating pumps are usually fitted with air vessels on the discharge side of the pump in order to reduce shocks and water hammer and produce smooth running and a uniform discharge. They are also fitted on the suction inlet to the pump when the suction pipe is

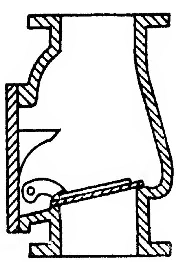


FIG. 5.—NON-RETURN VALVE

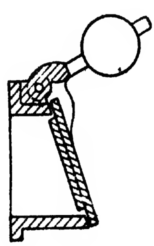


FIG. 6.—DELIVERY VALVE

long or of large area. The crank-driven pump, and particularly the single-acting type, requires an air vessel because the speed of the plunger varies in all parts of the stroke. The speed of the direct-acting pump does not vary to any great extent, but the rate of flow varies at each end of the stroke when the plunger comes to rest.

The internal diameter of an air vessel should not be less than the diameter of the pump plunger or piston and its height not less than the square root of its internal diameter multiplied by the water pressure in pounds per square inch divided by 15. Thus, if the internal diameter of the air vessel is 9 in. and the water pressure 100 lb., then $\sqrt{(9)}100/15 = 20$ in. minimum height.

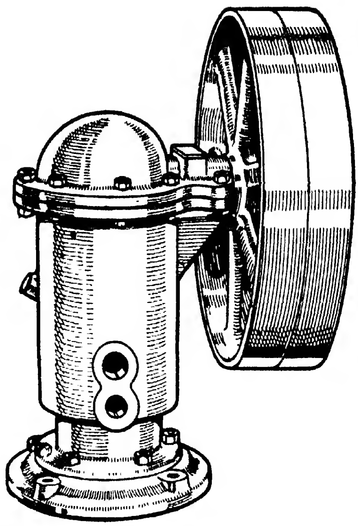


FIG. 7.—PLUNGER-PUMP EXHAUSTER
(*Ruston & Hornsby, Ltd.*)

Air vessels should be fitted as close to the valve box as possible, and when water hammer is experienced in spite of an air vessel, it is desirable to fit a gauge glass in order that a constant water height can be maintained with a cushion of air also of constant height. Usually a small valve on the suction side of the pump opening to atmosphere can be used to charge the air vessels, but on large pumps with long discharge pipes air pressure sufficient to produce smooth working is maintained by means of an air compressor. Air vessels on deep well pumps can also be automatically charged by a small air pump worked by the rise and fall of a water piston acting in conjunction with the pump plungers.

WORK DONE BY THE PUMP

Static Suction Lift

This neglects friction, and can be taken as the vertical height from the water-level to the centre of the pump, or the vertical height through which the liquid has to be raised.

Static Delivery Head

Static delivery head is the vertical height from the centre of the pump to the delivery water-level as in *A* and *B*, Fig. 8, or to the highest point in the delivery pipe when the discharge is arranged as at *C*.

Total Static Head

The total static head against the pump is the vertical height from water-level to water-level or from water-level to the highest point in the delivery pipe.

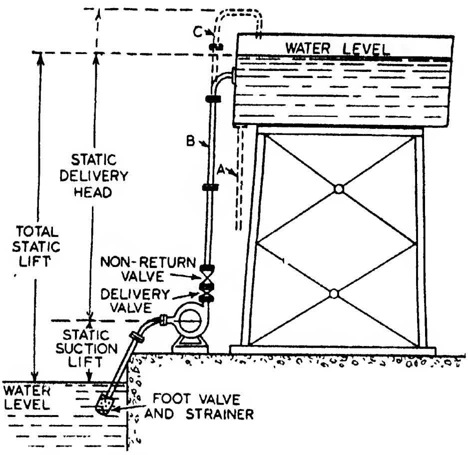


FIG. 8.—DIAGRAM SHOWING STATIC HEAD IN PUMP

Actual Head

The actual or manometric head is the total static head plus the losses due to friction in the pipes, fittings, and pump.

Friction Losses

The sum of these losses depends upon the sectional area and the internal condition of the pipes and fittings, the velocity and viscosity of the liquid being pumped, and the friction caused by bends, valves, and other fittings. Frictional resistance to the flow of water varies approximately as the square of the velocity. Table I gives the head in feet required to overcome the friction of flow in every 100 ft. of new straight cast-iron pipe for given quantities per minute. To these figures must be added the friction due to bends, valves, and other fittings from Table II. It will be seen that the greatest losses are caused by the velocity flow into a suction pipe not provided with a bell-mouthed entry and by T-pieces and elbows.

The following example shows the method of calculating the actual or manometric head against a pump. Let the velocity of the water be 350 gallons per minute and the total static lift 60 ft. The suction consists of a strainer with bell-mouthed entry, a foot valve, 6 ft. of straight 5-in. diameter pipe, and a 5-in. bend connected to the pump. Then:

SUCTION LOSSES

	Ft.
Strainer equivalent length	1·9
Foot valve equivalent length	4·7
Straight pipe equivalent length	6·0
Bend equivalent length	14·0
Total equivalent length	26·6

From Table I the equivalent head loss for 100 ft. of 5-in. pipe and 350 gallons per minute is 4 ft.; therefore, $4 \times 26{\cdot}6 \div 100 = 1{\cdot}06$ ft. head.

DELIVERY LOSSES

The delivery consists of a sluice or delivery valve, a non-return valve, 50 ft. of straight 5-in. pipe, and one 5-in. bend. Then:

	Ft.
Velocity head bell-mouth entry equivalent length . .	17·0
Delivery valve equivalent length	4·7
Non-return valve equivalent length	6·1
Straight pipe equivalent length	50·0
Bend equivalent length	14·0
Total equivalent length	91·8

Then delivery loss $4 \times 91{\cdot}8 \div 100 = 3{\cdot}67$ ft. head, and total frictional loss of pipes and fittings $1{\cdot}06 + 3{\cdot}67 = 4{\cdot}73$ ft. head. To this should be added a margin of, say, 25 per cent. for future deterioration of the pipes, making the loss $4{\cdot}73 + 1{\cdot}25 = 5{\cdot}98$ ft. head.

If the total static head is 60 ft., then the actual head against the pump will be $60 + 5{\cdot}9 = 65{\cdot}9$ ft. Except with short pipes of large diameter, it is usually a good investment to install all suction and discharge pipes of centrifugal pumps one standard size larger than the inlet and outlet connections on the pump. The reason for this will be seen from the table, where the loss or head caused by 250 gallons of water per minute flowing through 100 ft. of 2½-in. pipe is 70 ft. and the same quantity flowing through a 6-in. diameter pipe is only 0·9 ft.

The pressure in pounds per square inch due to 1 ft. head of water is 0·433 lb.; therefore the actual head in feet multiplied by 0·433 gives the head or pressure in pounds. A head of 2·3 ft. is equivalent to 1 lb. per square inch; therefore head in feet divided by 2·3 will give the head in pounds, and pounds per square inch $\div$ 0·433 is the head in feet.

Suction Lift

The theoretical suction lift of a pump at sea-level with water at 60° F. is $14{\cdot}7 \times 2{\cdot}3 = 33{\cdot}8$ ft., where 14·7 is the barometric pressure in pounds and 2·3 the water equivalent of 1 lb. per square inch. In practice actual lift depends upon the type and condition of the pump and the total length of the suction pipe, which may have a horizontal portion considerably longer than the vertical height. A plunger pump in good working condition will lift 23 ft. without difficulty, but it is advisable to limit the lift, if possible, to 20 ft. Centrifugal pumps can be constructed to lift up to about 20 ft., but normally the lift should not exceed about 15 ft.

For liquids other than water the theoretical suction life will be the equivalent of water divided by the specific gravity.

Temperature of Water

When the temperature of the water exceeds about 175° F., it should flow by gravity to the pump suction. This is on account of the vapour given off by the liquid counteracting atmospheric pressure on the surface of the water.

The following table gives the suction lift and head for various water temperatures:

130° F. 10 ft. suction lift; 190° F. 5 ft. head on suction valve.
150° F. 7 ft. suction lift; 200° F. 10 ft. head on suction valve.
170° F. 2 ft. suction lift; 210° F. 15 ft. head on suction valve.
175° F. 0 ft. suction lift; 220° F. 22 ft. head on suction valve.

TABLE I.—HEAD LOSS IN FEET FOR EACH 100 FT. OF STRAIGHT PIPE

Gallons per minute	Size of Pipe in Inches 1	1·5	2	2·5	3·0	4	5	6
10	13·3	1·7	0·4	—	1·0	—	—	—
20	51·0	6·5	1·6	0·5	—	—	—	—
30	—	14·8	3·5	1·1	0·47	—	—	—
40	—	26·0	6·0	2·0	0·8	—	—	—
50	—	40·0	9·4	3·0	2·15	—	—	—
60	—	57·0	13·5	4·3	1·8	0·4	—	—
70	—	78·0	18·0	5·8	2·4	0·55	—	—
80	—	—	24·0	7·5	3·1	0·71	—	—
90	—	—	30·0	9·4	3·9	0·9	0·29	—
100	—	—	37·0	12·0	4·0	1·12	0·36	0·16
120	—	—	51·0	16·5	6·8	1·6	0·5	0·22
140	—	—	69·0	22·0	9·1	2·12	0·66	0·3
160	—	—	—	29·0	12·0	2·75	0·87	0·4
180	—	—	—	36·0	14·9	3·45	1·1	0·5
200	—	—	—	45·0	18·3	4·25	1·33	0·6
250	—	—	—	70·0	27·5	6·45	2·05	0·9
300	—	—	—	—	40·0	9·3	3·0	1·3
350	—	—	—	—	53·0	12·5	4·0	1·75
400	—	—	—	—	—	16·0	5·2	2·3
450	—	—	—	—	—	20·0	6·5	2·8
500	—	—	—	—	—	25·0	8·0	3·4
600	—	—	—	—	—	34·0	11·3	4·8
700	—	—	—	—	—	—	15·4	6·5
800	—	—	—	—	—	—	19·6	8·3

TABLE II.—LOSS FOR FITTINGS IN EQUIVALENT LENGTHS OF STRAIGHT PIPE

Size of Pipe in Inches	Velocity Head for Ordinary Pipe Entry	Velocity Head for Bell-mouthed Entry	Bend	Foot Valve	Non-return Valve	Delivery Valve Full Open	Strainer	Ts and Elbows
1·0	4·5	2·7	2·5	0·8	1·0	0·8	0·3	2·7
1·5	7·1	4·3	3·7	1·2	1·6	1·2	0·5	4·3
2·0	9·8	5·9	5·0	1·7	2·2	1·7	0·7	5·9
2·5	12·6	7·6	6·4	2·1	2·8	2·1	0·9	7·6
3·0	15·6	9·4	8·0	2·6	3·5	2·6	1·0	9·4
4·0	21·0	13·0	10·7	3·6	4·7	3·6	1·4	13·0
5·0	28·0	17·0	14·0	4·7	6·1	4·7	1·9	17·0
6·0	35·0	21·0	17·3	5·8	7·6	5·8	2·3	21·0

CAPACITY OF RECIPROCATING PUMPS

The single-acting pump usually discharges water on its backward stroke though the pump cylinder, and its theoretical displacement will be equal to the sectional area of the plunger or piston multiplied by the length of stroke. Its theoretical capacity is therefore equal to the displacement multiplied by the number of strokes in a given period. This can be found by multiplying the area in square inches by the length of stroke in inches by the number of strokes per

minute, and dividing by 277·27, giving the capacity in gallons per minute, where 277·27 cub. in. equals 1 gallon of water.

For practical purposes the capacity of a single-acting pump can be found from the formula:

$$\frac{D^2 \times S \times N}{360} = G,$$

where G = gallons per minute.
D = diameter of plunger in inches.
S = length of stroke in inches.
N = number of strokes per minute.

Example.—Diameter of plunger, 6 in.; length of stroke, 12 in.; number of strokes per minute, 36.

Then: $$\frac{6 \times 6 \times 12 \times 36}{360} = 43 \text{ gallons per minute.}$$

The result obtained by this formula is about 3 per cent. below the theoretical displacement, but, owing to possible leakage past the piston or valves, or slip past the suction valve, is still about 3 per cent. greater than the actual discharge capacity under the best possible conditions.

Double-acting Pump

The double-acting pump discharges water on both the forward and backward stroke. On the forward stroke the effective area of the piston is reduced by an amount equivalent to the area of the piston rod. If this is neglected, then the double-acting pump displaces twice the amount of water of a single-acting pump of equal size.

The Duplex Pump

This consists of two pumps combined to form a single unit; the theoretical displacement is twice that of a simplex, or single-cylinder, pump of equal size.

The theoretical capacity of a double-acting duplex pump can be found from the formula:

$$\frac{D^2 \times S \times N \times 4}{6} = \text{gallons per hour,}$$

where D is the diameter of the water piston in inches, S the stroke in inches, and N the number of double strokes per minute.

Example.—Diameter of water piston, 7 in.; length of stroke, 10 in., with 35 double strokes per minute.

Then: $$\frac{7 \times 7 \times 10 \times 35 \times 4}{6} = 11{,}433 \text{ gallons per hour.}$$

Effective Area of Pump Piston

The displacement of a double-acting reciprocating pump is less on the forward or upward stroke on account of the pump rod. To find the average

effective area, let the diameter of the pump piston equal 8·5 in. and the rod 1·5 in. Then, if A is the area of the piston and a the area of the rod, $A - \frac{a}{2}$ = average effective area.

Area of $A = 8{\cdot}5 \times 8{\cdot}5 \times 0{\cdot}7854 = 56{\cdot}7$ sq. in.; and area of $a = 1{\cdot}5 \times 1{\cdot}5 \times 0{\cdot}7854 = 1{\cdot}7$ sq. in. $\therefore$ Average effective area $= 56{\cdot}7 - (1{\cdot}7 \div 2) = 55{\cdot}8$ sq. in.

Water Horse-power

The water horse-power can be found from equations:

$$\frac{G \times H}{3300} = W.H.P.$$

$$\frac{Q \times H}{8{\cdot}828} = W.H.P.$$

where G = gallons discharged per minute.
Q = cubic feet discharged per second.
H = actual head in feet.
$W.H.P.$ = water horse-power.

Efficiency of Pumps

The efficiency of a pump is the ratio of the water horse-power output to the brake horse-power, or:

$$\frac{W.H.P.}{B.H.P.} \times 100 = \text{percentage efficiency.}$$

Mechanical efficiency is the ratio $\frac{B.H.P.}{I.H.P.}$.

The efficiency varies considerably with design, size, and working condition of the pump. Direct-acting pumps on full load with strokes up to about 10 in. would probably average about 60 per cent. Larger pumps may rise as high as 80 per cent.

Power pumps with flywheel, gears, and cranks in first-class working condition may have an efficiency as high as 85 per cent. or even 90 per cent.

Standard centrifugal pumps normally average about 60 per cent. efficiency, but large pumps working under appropriate conditions will average about 80 per cent.

Small pumps must always be less efficient than larger ones of the same type, on account of the frictional horse-power being proportionally greater.

Power to drive a Pump

The brake horse-power to drive a pump can be found from the equation:

$$\frac{G \times H \times 100}{3300 \times E} = B.H.P.$$

where G = gallons discharged per minute.
H = actual head in feet.
E = efficiency.

Example.—Find the brake horse-power to pump 230 gallons per minute against a total head of 390 ft. Efficiency of pump 65 per cent.

Then: $$\frac{230 \times 390 \times 100}{3300 \times 65} = 41{\cdot}8\ B.H.P.$$

Steam Pressure to drive Pump

A pump has a steam piston 9½ in. diameter and a water piston 8 in. diameter. Its efficiency is 60 per cent., and it is run non-condensing with a back pressure of 5 lb. per square inch. What steam pressure would be required to pump against a total head equivalent to 125 lb. per square inch ?

Then: Pressure against pump $8 \times 8 \times 0{\cdot}7854 \times 125 = 6283$ lb.
Efficiency 60 per cent. $6283 \times 100 \div 60 = 10{,}471$ lb.
Pressure per square inch against steam piston:

$$\frac{10471}{9{\cdot}5 \times 9{\cdot}5 \times 0{\cdot}7854} = 147{\cdot}7 \text{ lb. per square inch.}$$

Assuming a loss of 12 lb. steam pressure with 5 lb. back pressure, then $147{\cdot}7 + 12 + 5 = 164{\cdot}7$ lb., say 165 lb., boiler pressure would be required.

Brake Horse-power absorbed by Pump

$$B.H.P. = W.H.P. \times \frac{100}{E}$$

Velocity of Water in Feet per Second

$$V = \frac{G}{2{\cdot}6 \times A};\ G = V \times 2{\cdot}6 \times A;\ A = \frac{G}{2{\cdot}6\ V}$$

where A = area of pipe in square inches.
G = gallons per minute.
V = velocity in feet per second.

DIRECT-ACTING PUMPS

The steam-driven direct-acting pump, in which the pump-plunger rod is a continuation of piston rod and no rotary motion of any kind is employed, is still largely used for pumping limited quantities of water against pressures up to about 250 lb. per square inch.

Horizontal Duplex Pump

The horizontal type of pump is to some extent easier to overhaul than the vertical, as the water piston can be withdrawn without disturbing the valve gear and piston rod, and bearings and working parts receive better lubrication; on the other hand, more floor space is occupied and cylinders are liable to greater wear on account of the weight of the pistons.

The steam consumption of the simplex or single-cylinder pumps of both horizontal and vertical types is usually less than with the duplex, but the latter gives a more continuous discharge on account of one piston commencing to discharge before the other has completed its stroke.

A horizontal duplex pump is shown in section in Fig. 9, and represents the Worthington-Simpson general-service type. It is constructed in fourteen standard sizes, the smallest having a water cylinder of $1\frac{1}{2}$-in. bore and 3-in. stroke and a water capacity of 200 gallons per hour, and the largest 7 in. by 10 in. with a capacity of 11,000 gallons per hour.

In the two smaller sizes the steam end is cast integral with the liquid cylinders. In all other sizes the water cylinder is a separate casting.

The steam valves are of the slide-valve type operated by direct lever connections.

Piston rods are usually of bronze, one-piece construction secured to the pistons by taper shanks and nuts.

The liquid cylinders and heads are of cast iron, with renewable bronze liners, but for special conditions all iron or all bronze is used.

The valves are all above the cylinders and the pistons submerged. They are of poppet type, bronze disc working on one-piece bronze guards and held on their seats by coiled bronze springs. Valve seatings are of bronze screwed into place on taper threads of fine pitch. When all iron cylinders are used, the valves and seatings are of iron with steel springs.

Water pistons are of bronze with removable followers, which allows adjustment and renewal of packing without removing pistons from the cylinders.

Setting Valves on Duplex Pumps

Valves are adjusted so that each valve is exactly central over its steam ports when the piston rod that operates it is in the centre of its stroke.

To find the centre of the stroke, push the piston rod hard against a cylinder cover and mark the rod, then against the opposite cover and make another mark. When a point midway between these two marks is in line with the face of the gland nut the rod will be in the centre of its stroke.

When both pistons are in the central position, the crossheads on the rods should be adjusted so that the levers for operating the valve rods are in a vertical position. The slide valves are then placed over the centre of the steam ports and adjusted so that the clearance is equally divided on either side of the valves.

The Weir Direct-acting Pump

The Weir direct-acting pump and its various parts are shown in Figs. 10 and 11.

The Slide-valve Chest

This is shown in section in Figs. 12–14. The chest contains a main and an auxiliary valve. The main valve *A* distributes steam to the cylinder. The auxiliary valve *B* has two functions: it distributes steam to work the main

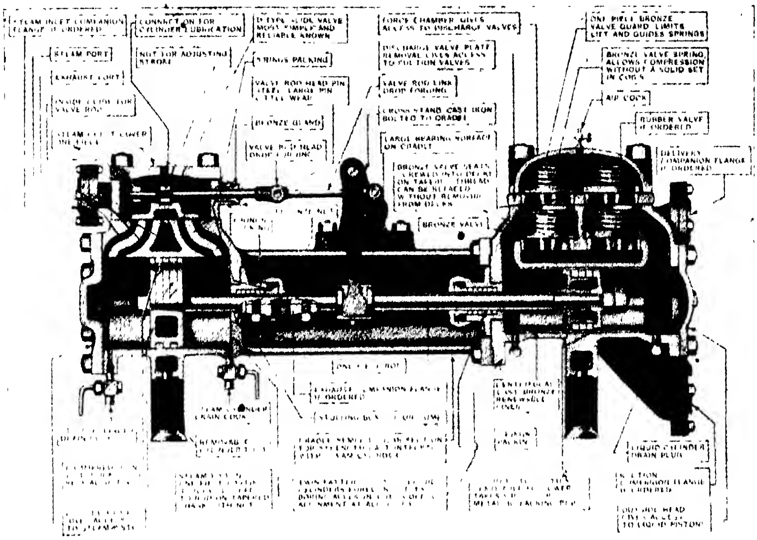


FIG. 9.—HORIZONTAL DUPLEX PUMP (*Worthington-Simpson, Ltd.*)

valve; and its outer edge cuts off steam entering the main ports *C* and *D* leading to the top and bottom of the cylinder.

Both valves are simple slide valves; but the main valve is half-round, and the round side works on the chest face, which is bored to fit. On the back of the main valve a flat face is formed upon which the auxiliary valve works. On this face the ports *C*, *D*, *E*, and *F* are cut with the exhaust ports *H* in the centre (Fig. 17). The ends of the main valve are cylindrical, and project beyond the port face; they are fitted with loose bells or cylinders in which the valve works. These bells are held in position by the end covers and by faces cast on the chest for the purpose.

The operation of the valves during a double stroke of the pump is as follows:

When the piston is at the bottom of the stroke, the main valve is in the right-hand position, as in Fig. 13, the auxiliary valve *B* is also at the bottom of its travel, and the port *C* leading to the bottom of the cylinder is open to the steam pressure. This port remains open until the piston reaches half-stroke, when the auxiliary valve begins to move in the same direction as the piston, and at about three-quarters stroke the auxiliary valve closes the port *C* leading to the bottom of the cylinder. The remainder of the stroke is completed by the expansion of steam already shut in the cylinder, or by more steam admitted through the by-pass, which will be described later. Port *E* leads to the left-hand end of the main valve, and port *F* to the right-hand end. When the piston reaches

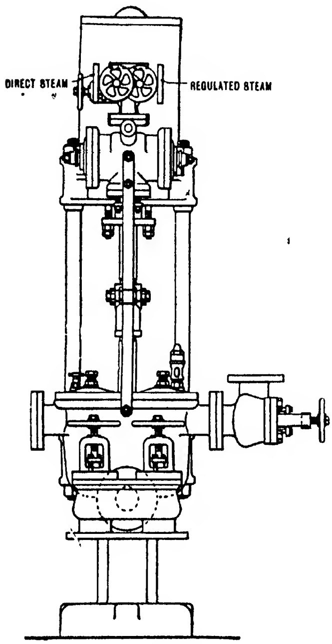


FIG. 10.—DIRECT-ACTING FEED PUMP
(*G. & J. Weir, Ltd.*)

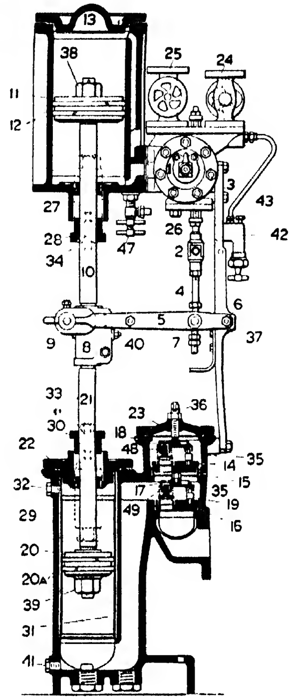


FIG. 11.—BOILER FEED PUMP
(*G. & J. Weir, Ltd.*)

FIG. 11.—2. Double joint. 3. Front stay. 4. Bottom spindle. 5. Valve-gear levers. 6. Front-stay bush. 7. Ball crosshead. 8. Main crosshead. 9. Crosshead pin. 10. Piston rod. 11. Piston body. 12. Piston rings. 13. Cylinder cover. 14. Discharge-valve seat. 15. Discharge-valve seat ring. 16. Suction-valve seat. 17. Suction-valve guard. 18. Discharge-valve guard. 19. Water valves. 20. Bucket. 20A. Bucket rings. 21. Pump rod. 22. Pump cover. 23. Valve-chest cover. 24. Steam stop valve. 25. Exhaust stop valve. 26. Auxiliary-valve spindle. 27. Cylinder neck ring. 28. Cylinder gland. 29. Pump neck ring. 30. Pump gland. 31. Pump liner. 32. Liner fixing pin. 33. Pump-gland studs. 34. Cylinder-gland studs. 35. Suction and discharge columns. 36. Top pins. 37. Ball crosshead bushes. 38. Piston-rod nut. 39. Pump-rod nut. 40. Crosshead taper pin. 41. Pump drain plug. 42. Plunger lubricator. 43. Adapter for lubricator. 45. Gland packings complete. 46. Pump and cylinder joints. 47. Cylinder drain valve. 48. Discharge-valve spring. 49. Suction-valve spring.

the top of its stroke, port *E* is opened to the exhaust by the auxiliary valve *B*, and at this position port *F* is open to steam; the main valve is therefore thrown over until the exhaust from the left-hand end is cut off; the steam remaining after cut-off acts as a cushion and prevents the main valve hitting the end cover. The main valve is now at the opposite end of its stroke, and the port *C*, which

admits steam to move the piston on the up stroke, is open to exhaust. The port *D* leading to the top of the piston is now open to steam.

The action described above also takes place on the down stroke; the main piston moves half its stroke before beginning to move the auxiliary slide, which again cuts off steam at about three-quarters of the stroke. The remainder of the stroke is travelled by the expansion of the steam in the cylinder, or by fresh steam admitted through the by-pass.

By-passes

Under certain conditions the pump will not complete its stroke by the expansion of steam in the cylinder; for instance, if the pump were started with the cylinder cold, the steam would rapidly condense and fall below the pressure necessary to move this piston. In such circumstances it is necessary to admit steam after the auxiliary valve is closed to main ports *C* and *D* on the face of the main valve, and this is done by means of the by-pass on each end of the chest. These by-passes *I* and *J* (Figs. 12 and 16) are made by cutting a port on the back of the main valve, and provision is made on each end of the steam chest for opening and closing the ports by hand.

Pump Bucket

The bucket *A*, Fig. 18, is made of gunmetal, in which two grooves are turned. Rings *BB*, of specially manufactured ebonite, are fitted into these grooves, and are cut at an angle *CC* to permit of their expanding and filling the pump chamber.

In order to fit new rings a ring is first inserted in a groove to test lateral clearance, which should be 0·01 in. for pumps up to 6-in. bore and 0·015 in. for pumps above that size. The ring is then moved until it has passed circumferentially completely within the groove. It is then sawn through at the angle shown, with a fine hacksaw, so that the gap is approximately $\frac{1}{32}$ in.

When fitting over size rings or when dealing with temperatures above 212° F. the gap should be adjusted by filing to give the following clearances, with the ring in position in the pump cylinder. for pumps up to 6-in. bore, $\frac{1}{32}$-in. gap.; pumps 7-in. to 9-in. bore, $\frac{1}{16}$-in. gap, and for pumps of 10-in. bore and upwards, $\frac{3}{32}$-in. gap.

The ring, being now ready for springing into the groove, must be completely submerged in boiling water until rendered pliable. Then, working from the point opposite the saw-cut, press ring into groove. When ring is pressed home, it should have a saw-cut $\frac{3}{8}$ in. open. The ring is held in this position for about one minute until it sets hard. When the bucket, fitted with the hardened ring and a gap of $\frac{3}{8}$ in., is dropped into the pump chamber, the gap again closes to the proper size, thus giving the necessary compression on the ring to ensure freedom from leakage.

Adjusting Length of Stroke

In order to adjust the length of stroke, a pair of dividers should be set to the distance *X* indicated by the centre dots on the front of the stay, Fig. 19; then

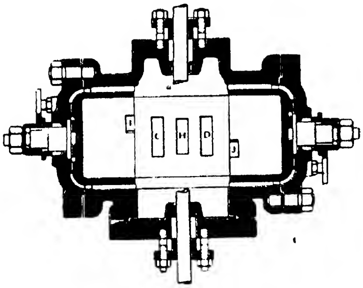


FIG. 12.—SECTIONAL FRONT ELEVATION OF VALVE CHEST

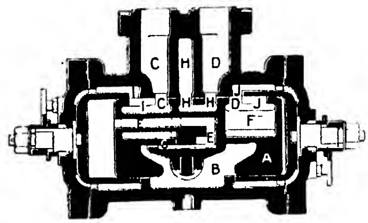


FIG. 13.—SECTIONAL PLAN OF VALVE CHEST

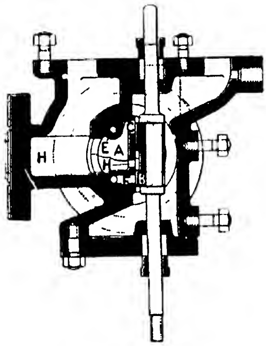


FIG. 14.—SECTIONAL SIDE ELEVATION

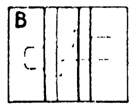


FIG. 15.—AUXILIARY VALVE

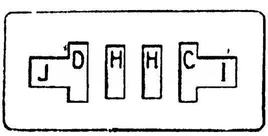


FIG. 16.—MAIN VALVE FACE

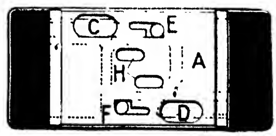


FIG. 17.—MAIN VALVE BACK—FACED FOR AUXILIARY VALVE

adjust the valve spindle and nuts so that the centre dots on these correspond with the points of the dividers as shown in the illustration.

An alternative method is as follows: while the pump is working on normal duty, slacken the locknut *A* and gradually screw up the bottom portion of the valve spindle *B* by means of the hexagon nut *C* until the piston strikes the cylinder cover. Then measure the distance between the steam-cylinder gland and the top of the crosshead *D*. Next screw the spindle *B* in a downward direction

until the distance between the gland and the crosshead is ½ in. more than when previously measured. This completes the operation for the top part of the stroke; the process should then be repeated for the bottom end of the stroke.

The Pearn Vertical Direct-acting Pump

The steam end of this simplex type of pump is illustrated in Fig. 20. The steam distribution is as follows:

The piston being at the bottom of the stroke, steam is admitted to the cylinder through the main port 1 to the by-pass port 2 below the piston, thus slowly raising it until the main port 1 is uncovered to full steam. On the piston approaching the end of its stroke the main port 3 is covered by the piston and the remaining steam exhausted slowly through the by-pass port 4, bringing the piston gradually to rest. At this moment the auxiliary slide valve 5 (mechanically controlled from the crosshead on the pump rod), arriving at the end of its downward stroke, opens the steam port 6 to the top of the auxiliary piston 7 and connects steam port 8 to the exhaust port 12. The auxiliary piston 7 then moves downwards, carrying with it the main slide valve 9, thus opening the main port 3 to the top of the steam piston through the by-pass port 4. A reversal of the above performance completes the down stroke of the piston.

From the sectional view through the auxiliary slide valve, it will be seen that the steam and exhaust ports to the auxiliary cylinder are arranged left and right at the top and right and left at the bottom, and that the exhaust port 12 is always open, thus ensuring a free escape for the steam from the auxiliary piston when the exhaust ports 8 and 11 are opened.

The motion of the auxiliary slide valve is continuous, except for the small amount of lost motion between the nuts *A* and *B*, which are for the purpose of adjusting the stroke of the pump.

Pump Barrel and Valve Box

The pump barrel and valve box are shown in Fig. 21. The former is designed without long passages, so that it can be easily cleaned when the water has a depository nature. The liner is of gunmetal forced into the barrel with a special form of joint made at *A*. It is extended at the upper end to the level of the cover, by which the liner is securely held in position, no outside screws being necessary.

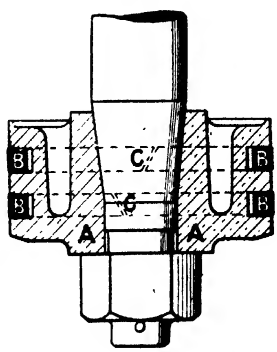


FIG. 18—PUMP BUCKET

Pumps up to and including 5 in. diameter have the barrels and valve boxes cast together. They are fitted with single valves of the mitre type with long guides and bronze springs. The valve seatings are tapered on the outside and pressed in as shown on the right of Fig. 21. Pumps 5½ in. diameter and upwards are fitted with group valves of mitre type, with long

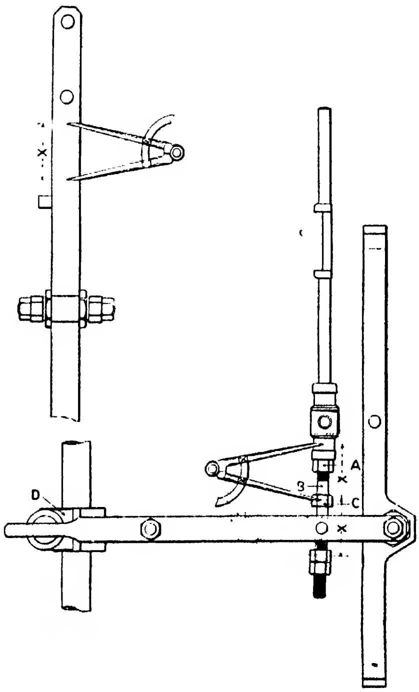


FIG. 19.—METHOD OF ADJUSTING LENGTH OF STROKE—WEIR DIRECT-ACTING PUMP

guides to reduce wear, and bronze springs. The valve seats are of gunmetal secured by manganese-bronze setscrews.

The packing glands for both steam pistons and pump rods are made in two parts, the flanges bearing on a convex surface on the bushes; thus any binding on the rods caused by screwing the glands down unevenly is obviated.

The Pump Piston

The pump piston used with this pump is shown in Fig. 22. It is of simple construction, and is fitted with one-piece packing rings made from flexible material of a composition which is tough and operates with little friction. The rings operate on the hydraulic principle, viz. by means of ports drilled in the piston, top and bottom, which admit liquid on the pressure strokes to expand them radially and uniformly against the walls of the liner; the lips of the ring are also forced against the plates between which they are mounted.

The rods are screwed into the crosshead and secured by a cotter and locknut. They are made sufficiently large in diameter to allow for returning when worn, without interfering with the threads at the end.

This type of pump is manufactured by Messrs. Frank Pearn & Co., Ltd., in sixteen standard sizes from 2½ in. diameter by 8 in. stroke with a displacement of 340 gallons per hour at 22 double strokes per minute, to 10½ in. diameter by 24 in. stroke and a displacement of 6,900 gallons per hour at 14 double strokes per minute.

Worthington Vertical Simplex Pump

This is shown in the sectional views in Fig. 23. The working parts of the steam end consist of a main slide valve 1, a steam-thrown plunger 3, and a small auxiliary slide valve 2. These three parts, together with the valve-rod connections 6 and 7 to the lever which moves the auxiliary valve 2, make up the whole working mechanism. It is particularly adapted for operation with high-pressure

steam, the auxiliary valve being the only working part directly connected to the valve rod. This valve, due to its small area, is subjected to but little friction from the pressure of steam upon it. It will be noted that the auxiliary cylinder carrying the steam-thrown plunger is at right angles to the length of the main cylinder, so that the weights of the plunger and main valve are supported, thus preventing any possibility of these parts dropping and short-stroking the pump.

In the sectional view of the steam cylinder on the left of Fig. 23 it will be seen that the main piston is at mid-stroke and moving downwards. The auxiliary valve, through the valve connections, has just been moved to the upper end of its stroke, admitting steam to the auxiliary cylinder 4, thus forcing auxiliary plunger 3 with its main valve 1 to the left-hand of its movement. The main valve has opened the steam passage to port leading to upper end of the main cylinder and exhaust passage leading to lower end. Steam is entering cylinder 5 through steam-pipe opening 25, filling the steam spaces of the auxiliary cylinder and upper end of the main cylinder, while at the same time exhaust steam from the lower end is passing out at the main exhaust opening.

The auxiliary plunger 3 is operated through two small ports by means of the auxiliary valve. This valve, riding upon the back of the main valve, admits steam to one end and exhausts from the other end of the steam-thrown plunger through passages 21 and 22 in the main valve face, and through exhaust passage 36 and cavity 30 leading to the main exhaust 13. As the plunger approaches the end of its travel, the exhaust port 36 is closed by the movement of the main

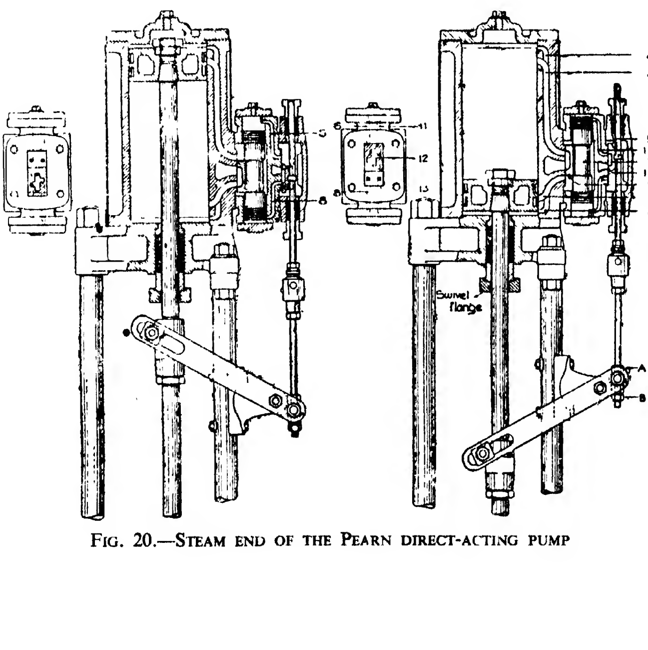


FIG. 20.—STEAM END OF THE PEARN DIRECT-ACTING PUMP

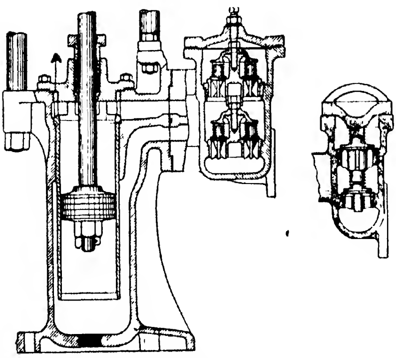


FIG. 21—PUMP BARREL AND VALVE BOXES

valve, the steam trapped in the cylinder end forming a cushion upon which the plunger comes to rest without shock and with the main valve in the correct position for the return stroke of the main piston. The main valve 1, by the auxiliary plunger, is carried across the valve seat of the main cylinder, thus controlling the admission and exhaust of steam to and from that cylinder.

This type of pump is manufactured by Messrs. Worthington-Simpson, Ltd., in seventeen standard sizes for boiler feeding, the smallest pump having a capacity of 72 gallons per hour at 40 double strokes per minute and the largest 15,000 gallons per hour at 14 double strokes per minute. For general service work ten standard sizes are made, the normal capacities ranging from 11,180 to 26,500 gallons per hour.

In the 7-in. × 5-in. × 12-in. size and upwards, the suction and delivery-valve seats are formed in plates, which plates are separate from the pump casting and are arranged with valves, guards, and springs on one spindle. By this grouping arrangement the whole of the suction and delivery valve service for each side of the pump forms a complete unit, and therefore can be taken out for renewal or inspection with a minimum of time and trouble.

POWER PUMPS

When a pump is driven by means of a motor, engine, belt, or ropes through the medium of gears and cranks, it is termed a power pump. It may be of horizontal or vertical type, with one or more water cylinders, and be either single- or double-acting.

The water discharged by a one-cylinder single-acting pump or even a duplex

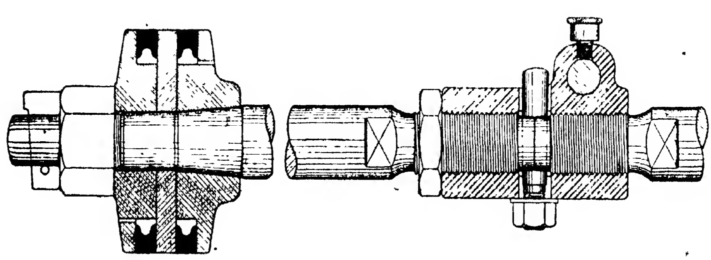


FIG. 22.—PUMP PISTON OF PEARN PUMP

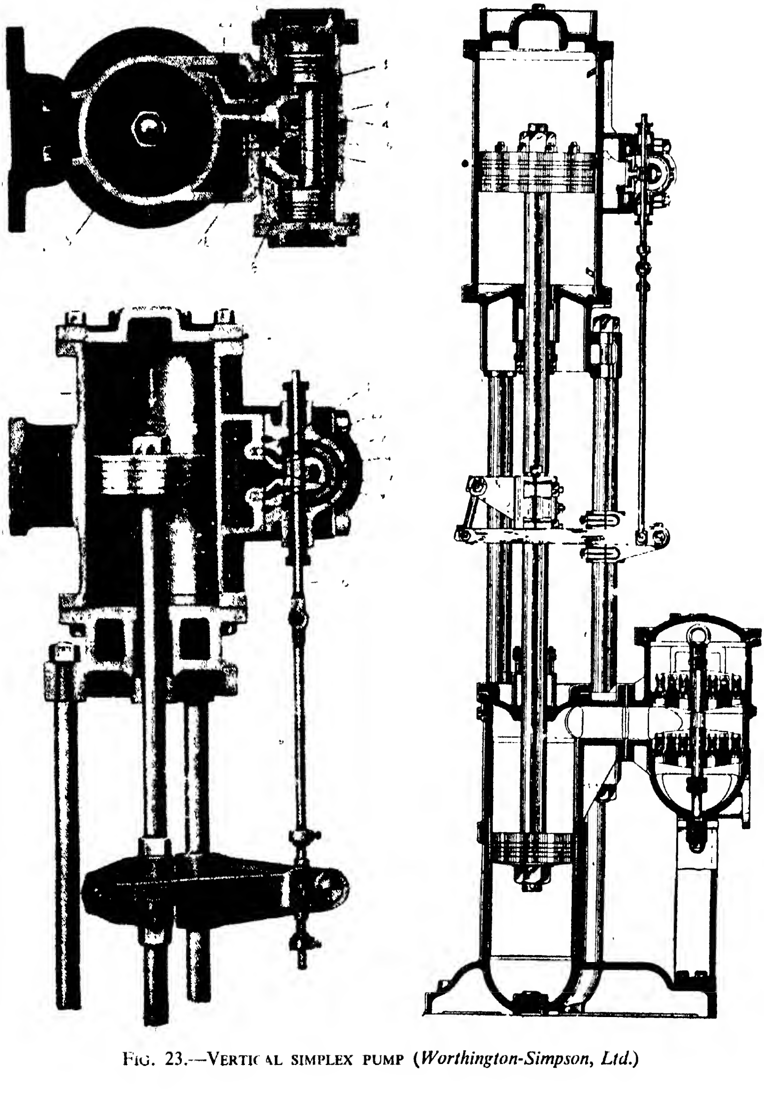


FIG. 23.—VERTICAL SIMPLEX PUMP (*Worthington-Simpson, Ltd.*)

single-acting pump must be intermittent and the flow from a triplex pump much more uniform. Three-throw pumps with cranks at 120° give a very smooth and uniform discharge, and any addition to the number of water cylinders would make little difference to the regularity of flow.

Vertical Triplex Ram Pump, Four-bearing Type

The triplex pump shown in Fig. 24 is constructed with either single- or double-reduction gears and for belt or motor drive. When arranged for direct driving the baseplate is extended and provided with a stool for the motor.

Side frames are of cast iron of H section, with strengthening ribs in the centre. Angular pedestals are fitted on the side frames with gunmetal bearings to carry the countershaft.

Two crankshaft pedestals are provided on the frames, and two inner pedestals are carried on the guide casting, reinforced by steel stanchions bolted to the baseplate.

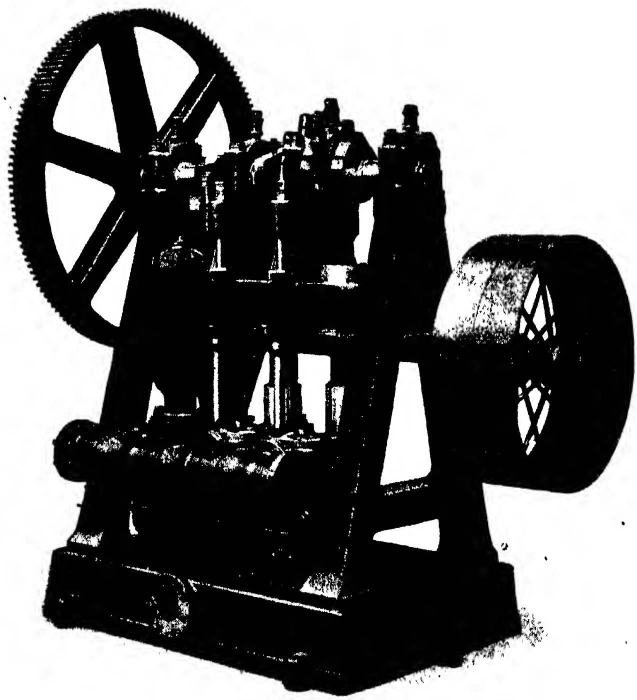


FIG. 24.—VERTICAL TRIPLEX RAM PUMP, FOUR-BEARING TYPE
(Lee, Howl & Co., Ltd.)

Single-reduction gearing has the machine-cut spur wheel and pinion 3–5 to 1 ratio. Double-reduction gearing has the motor pinion of rawhide engaging with the machine-cut spur wheel on the countershaft.

TABLE III.—TYPICAL RAM PUMP SIZES.

	Single Gearing and Double Gearing				
Diameter of rams in inches	5	6	7	8	8
Length of stroke in inches	6	8	8	8	12
Capacity in gallons per hour	3,820	7,330	9,960	13,000	17,585
Revolutions per minute of crankshaft	50	50	50	50	45
Diameter of suction and delivery in inches	3½	4½	5	6	7
B.H.P. for 600 ft. head	16·5	30	42	54	68

Pumps of this type are constructed for heads up to 2,000 ft. The figures given above are for heads up to 600 ft.

Horizontal Duplex Power Pump

A horizontal duplex enclosed pump is shown in Fig. 25. This pump is suitable for delivering 1,500 gallons per hour against 200 lb. per square inch,

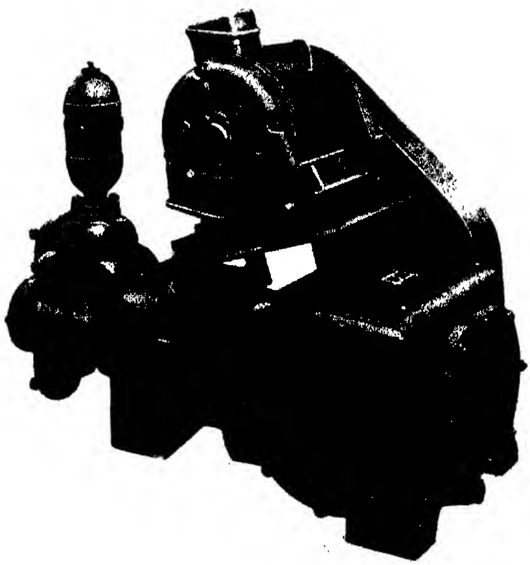


FIG. 25.—A HAYWARD-TYLER ENCLOSED DUPLEX PUMP

single-acting pump must be intermittent and the flow from a triplex pump much more uniform. Three-throw pumps with cranks at 120° give a very smooth and uniform discharge, and any addition to the number of water cylinders would make little difference to the regularity of flow.

Vertical Triplex Ram Pump, Four-bearing Type

The triplex pump shown in Fig. 24 is constructed with either single- or double-reduction gears and for belt or motor drive. When arranged for direct driving the baseplate is extended and provided with a stool for the motor.

Side frames are of cast iron of H section, with strengthening ribs in the centre. Angular pedestals are fitted on the side frames with gunmetal bearings to carry the countershaft.

Two crankshaft pedestals are provided on the frames, and two inner pedestals are carried on the guide casting, reinforced by steel stanchions bolted to the baseplate.

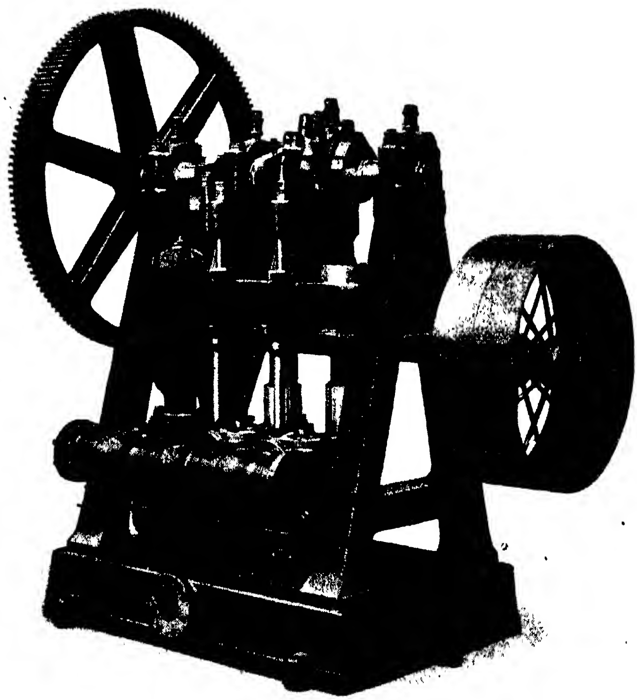


FIG. 24.—VERTICAL TRIPLEX RAM PUMP, FOUR-BEARING TYPE
(Lee, Howl & Co., Ltd.)

The engine, which operates on the four-stroke principle, has a large oil sump, and is provided with forced-feed lubrication. Pistons are of heat-resisting alloy, and the cylinders are fitted with detachable "wet-type" (water-cooled) liners of specially hardened material.

Outputs of these pumping sets range from 80 gallons per minute against a head of 120 ft. to 3,000 gallons per minute at 15 ft.

A typical belt-drive pump powered by a small diesel engine is illustrated in Fig. 26.

Hydraulic Pumps

Hydraulic pumps are used to supply water or other liquids under pressure in order to transmit power to presses, rams, and lifts, and for the many purposes for which hydraulic pressure is required in modern industry.

When large outputs are needed the direct-acting steam-driven reciprocating pump is frequently used, and for smaller capacities the power-driven three-throw ram type is more often adopted. The rotary pump has also been applied to this class of work with considerable success when the output is comparatively small.

Hydraulic pumps are often connected to a loaded ram or accumulator, which allows the pump to run more or less continuously when the water from the accumulator is being used intermittently.

A three-throw pump manufactured by Messrs. T. H. & J. Daniels, Ltd., is shown in Fig. 27, with a sectional arrangement in Fig. 28. It is designed primarily for use with an accumulator in the operation of hydraulic presses. The drive may be arranged either through a multiple V-rope drive or by a motor through reduction gearing. The former arrangement is shown with the sectional drawing and the latter in the external view.

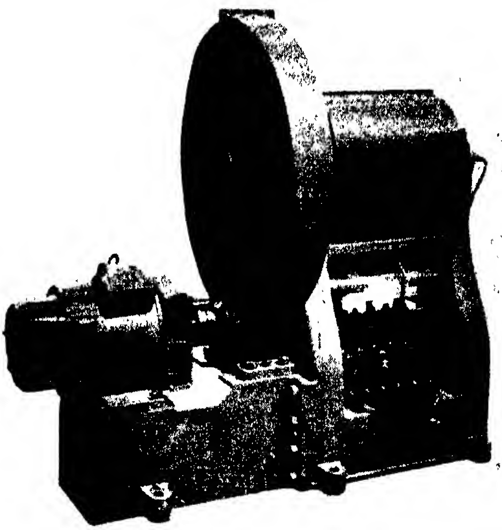


FIG. 27.—THREE-THROW HYDRAULIC PUMP (*T. H. & J. Daniels, Ltd.*)

The pump is intended for a crankshaft speed of 100 revolutions per minute and will deliver 5·5 gallons per minute

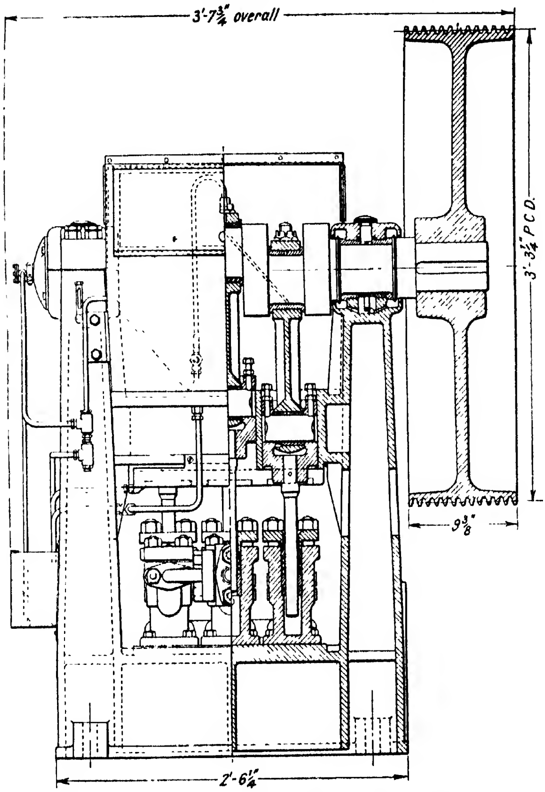


FIG. 28.—SECTIONAL ARRANGEMENT OF 1¼-IN. DIAMETER BY 3-IN. STROKE THREE-THROW HYDRAULIC PUMP
(*T. H. & J. Daniels, Ltd.*)

at 1 ton per square inch pressure. Other sizes give capacities from 2·5 gallons to 25 gallons per minute, all at 1 ton per square inch pressure. These smaller pumps can also be arranged to give 1½ tons per square inch pressure.

The frame, as will be seen, is a rigid one-piece casting. The pump bodies are bolted to the base cross-connections, and the cylindrical crossheads work in the upper cross-connection.

The crankshaft, which is machined from a steel forging, runs in large plain bronze bearings. This type of bearing is also used for the top and bottom ends of the connecting rods, except that the small-end bearings are of the bush type. The gudgeon pins are fixed in the crossheads. The main bearings are ring lubricated, the shaft having oil-throw-off collars.

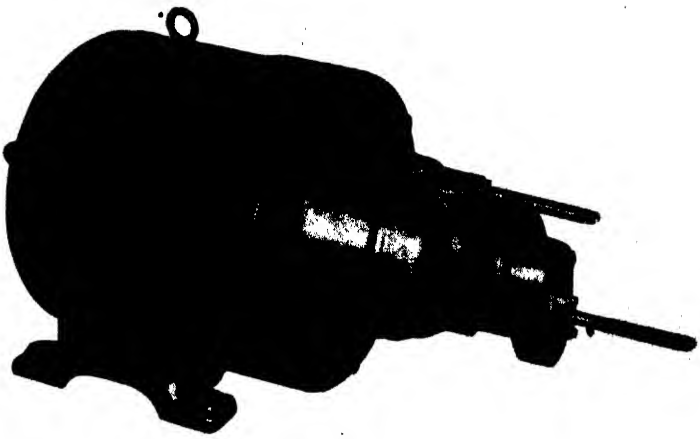


FIG. 29.—AN ELECTRAULIC AXIAL-PLUNGER PUMP
(*Towler Bros. (Patents), Ltd.*)

Lubrication for the other components is obtained from a small pump mounted on the crankshaft end cover, and a drip tray under the crosshead returns the oil to the suction tank.

The pump bodies are independent and are of cast steel. The plungers are of stainless steel and work in bronze liners and glands. Their diameter is 1·5 in. and the stroke 3 in.

The valves are of bronze and the seats of stainless steel. Three branch inlet and outlet manifolds are bolted to the bodies.

All renewable parts are readily accessible, and the crankshaft and flywheel are fitted with easily renewable guards.

High-pressure Electraulic Axial-plunger Pump

Another interesting type of high-pressure pump suitable for supplying hydraulic machinery is the "Electraulic" axial-plunger pump which is illustrated in Figs. 29 and 30. Fig. 29 shows the external view whilst the internal details are illustrated in Fig. 30.

CONSTRUCTION AND OPERATION.—The driving shaft (1) (Fig. 30) is a press fit in the swash member (3), and is supported in a roller-bearing journal (2), which is very lightly loaded, and whose sole function is to maintain the shaft and swash member in concentric alignment. The swash member (3) has an oblique face (3*a*) and a transverse face (3*b*). The end thrust of the swash member is taken by the film-lubricated thrust-washer (4) attached to the transverse face of the swash member. The wobble plate (5) is mounted on a short journal bearing (6) whose function is to maintain concentric alignment between the

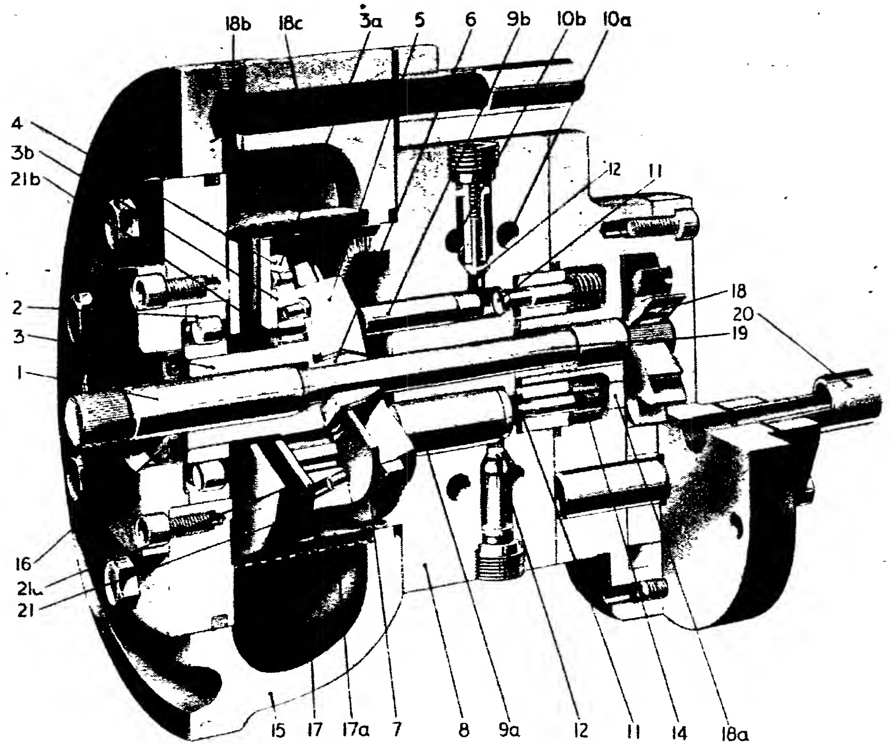


FIG. 30.—SECTIONAL VIEW OF "ELECTRAULIC" AXIAL PLUNGER PUMP

1. Driving shaft.
2. Roller-bearing journal.
3*a* & *b*. Swash member.
4. Thrust washer.
5. Wobble plate.
6. Journal bearing.
7. Thrust washer.
8. Pump body.
9*a* & *b*. Plungers.
10*a*. Discharge manifold.
10*b*. Delivery manifold.
11. Inlet valves.
12. Discharge valves.
14. Inlet manifold.
15. Swashplate casing.
16. Oil seal.
17 & 17*a*. Screens.
18. Booster pump.
18*a*, *b*, & *c*. Oil passages.
19. Shaft.
20. Inlet pipe.
21. Stationary plate.
21*a*. Stationary plate, radial holes.
21*b*. Swash member, axial passage.

(*Towler Bros.* (*Patents*), *Ltd.*)

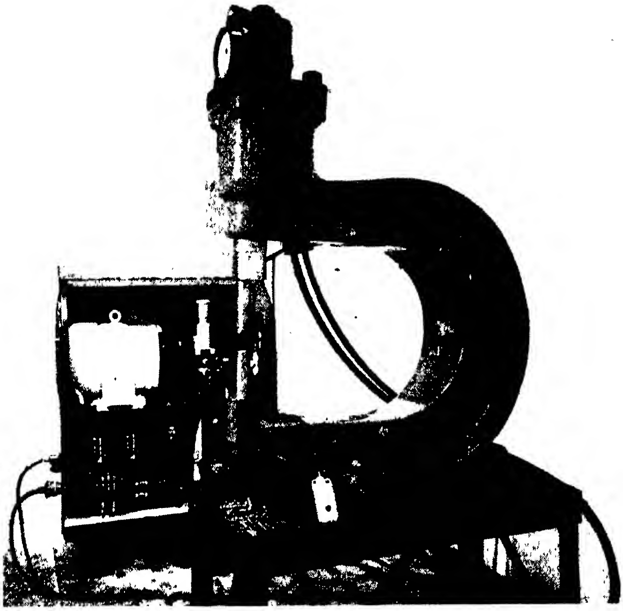


FIG. 31.—A 30-TON PORTABLE BEAR-TYPE RIVETER POWERED BY AN ELECTRAULIC TANK-MOUNTED AXIAL-PLUNGER PUMP

Working pressure is 2 tons per square inch. (*Towler Bros.* (*Patents*), *Ltd.*)

wobble plate and the swash member. The thrust of the wobble plate is taken by the film-lubricated thrust washer (7) attached to the oblique face of the swash member. The complete swash-plate mechanism is enclosed in a casing (15), one end of the casing being closed by the stationary pump body (8), the other end of the casing enclosing the driving shaft and being provided with an oilseal (16).

The booster pump (18), a simple gear pump, is driven by flexible shaft (19), which is an extension of the driving shaft (1). The hydraulic oil passes from the pump reservoir by the inlet pipe (20) to the booster pump, and thence by a passage (dotted) (18*a*) to the inlet manifold (14) of the high-pressure pump.

The oil within the screen is replenished by that which leaks past the pump plungers, and, although this leakage is minute, it is sufficient to renew the oil within the screen with reasonable frequency during the operation of the pump, thereby maintaining its lubricating quality.

The high-pressure pump body (8) is stationary, and contains six axial plungers, three large plungers (9*a*), and three small plungers (9*b*), which are a fine lap fit in their respective bores. The inlet valves (11) and discharge valves

FIG. 32.—A NITROGEN-LOADED ACCUMULATOR UNIT COMPLETE WITH ELECTRAULIC PUMP AND TANK (*T. H. & J Daniels, Ltd.*)

(12), of the self-acting mushroom type, are situated at the inner end of each pump bore. The inlet valves are in communication with the inlet manifold (14) and the discharge valves of the large plungers, with discharge manifold (10*a*). The discharge valves of the small plungers are in communication with delivery manifold (10*b*).

It will be seen (Fig. 29) that the high-pressure pump plungers (9*a* and 9*b*) protrude from the pump body and the plunger ends make contact with the surface (5*a*) of the wobble plate. The plunger ends are specially shaped, spherical or quasiconical, so as to provide a large radius of contact between them and the surface of the wobble plate. The driving shaft and swash member (3) rotate, whereas the pump body (8) is stationary, but the wobble plate (5) is made to rotate very slowly in a reverse direction to the driving shaft by means of bevel gear, one wheel being attached to the pump body and the other wheel being attached to the wobble plate. This geared wobble plate, in combination with film-lubricated thrust washers, enables the pump to be operated continuously at very high pressures without appreciable wear. A prototype pump has been operated for over 2,000 hours at a *continuous* pressure of 7,000 lb. per square inch without any measurable sign of wear.

The combined unloading-relief valve (13) can be arranged to operate as a sustained pressure valve, in which case there is only a difference of 200–300 lb. per square inch between the predetermined pressure at which the large plungers are unloaded or by-passed and the maximum pressure at which pressure is sustained with the discharge of the small plungers blowing off at the relief valve.

Alternatively, the valve (13) may be arranged to operate the pump as a two-stage high-pressure pump, in which case there is a difference of 2,000 or 3,000 lb. per square inch between the predetermined pressure at which the large plungers are unloaded or by-passed and the maximum pressure at which the small plungers blow off at the relief valve. The booster pump (18) may be increased in capacity up to 10 gallons per minute (2,770 cub. in.) at 500 lb. per square inch, thereby providing low pressure for the rapid approach stroke of the press ram, with the addition of sustained pressure or two-stage high pressure for the pressing operation, as described above.

The unloading-relief valve is adjustable, so that the maximum blow-off pressure may be varied at will to suit the pressing operation.

CENTRIFUGAL PUMPS

Centrifugal pumps can be broadly divided into two classes, under the heading of volute pumps, in which a volute or spiral-shaped casing surrounds a rotating impeller, or guide-vane pumps, where the liquid being pumped leaves the tip of an impeller and enters the vanes of a diffuser. A combination of the two systems is sometimes adopted for particular duties, but generally the volute type is employed when the volume of liquid is large compared with the head and the guide-vane multi-stage pump, when the head is large in relation to the volume.

In all cases pressure energy is obtained by the rotation of an impeller. The liquid entering the eye or centre of the impeller is thrown outward by centrifugal force, and the kinetic energy developed is largely converted into pressure energy by the uniformly increasing area of the volute casing or by the diffusing vanes which prevent the dissipation of the kinetic energy.

The Volute Pump

The volute type of pump is largely used for irrigation, sewage, sand, gravel, circulation, and general-purpose pumping. An outline of this type is shown in Fig. 33. As will be seen, the impeller blades are curved backwards away from the direction of rotation to reduce shock, and the entry passages are designed to bring the liquid into the eye of the impeller with the minimum disturbance. The volute

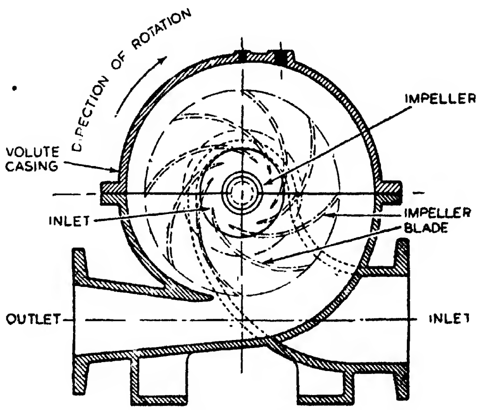


FIG. 33.—THE VOLUTE PUMP

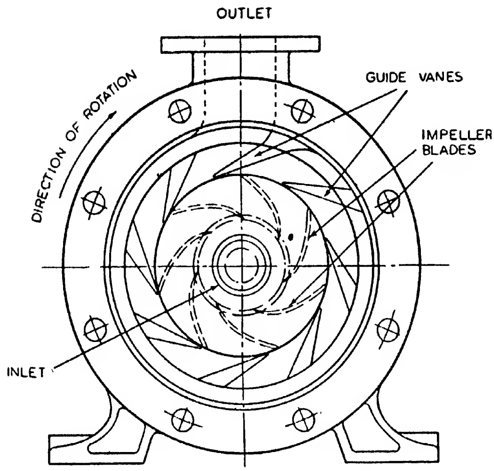


FIG. 34.—THE GUIDE-VANE PUMP

area has to be carefully proportioned, and also the relative size of the impeller eye to the diameter of its periphery.

Although the volute pump is simple in construction, so many variations can be made in the details of its design that specialised knowledge and a considerable amount of practical experience are definitely required of the successful designer.

The Guide-vane Pump

A section through a guide-vane pump manufactured by Messrs. Gwynnes Pumps, Ltd., is shown in Fig.34. This type is frequently termed a turbine pump, on account of its action being similar to a reversed reaction turbine.

When the total head against the pump exceeds about 200 ft., the speed of a volute pump becomes excessive, and for that reason the guide-vane pump with impellers in series is mainly adopted. Multi-stage pumps have been constructed to work against pressure up to 2,000 lb. per square inch.

Combined Volute and Guide-vane Pump

An example of combined volutes and guide-vane passages is shown in Fig. 35. This type was designed and used in the single-stage Gwynne pressure pump shown in Fig. 36. This pump gives an output of 17,600 gallons per minute against a total gauge head of 164 ft., at 750 revolutions per minute, and with 83 per cent. efficiency.

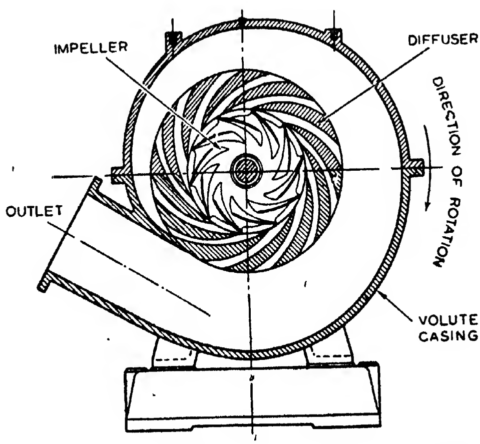


FIG. 35.—COMBINATION OF VOLUTES AND GUIDE-VANE PASSAGES (*Gwynnes Pumps, Ltd.*)

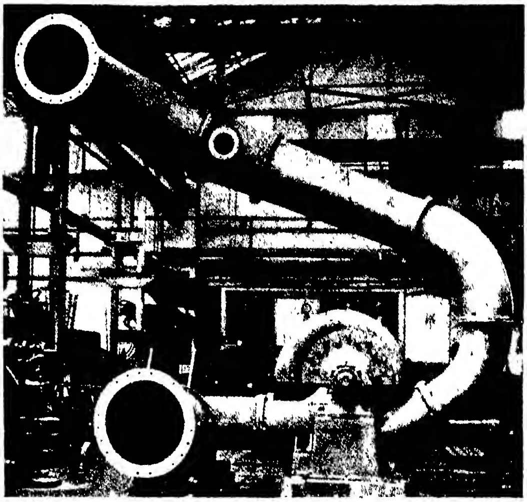


FIG. 36.—A SINGLE-STAGE GWYNNE PUMP
Output, 17,600 gallons per minute.

Pump Inlet Passages

These take many forms, and may be single or double entry. When guide vanes are omitted, the liquid passes directly into the whirling chamber of the volute casing through a filling ring. A section through a double-flow volute pump is shown in Fig. 37.

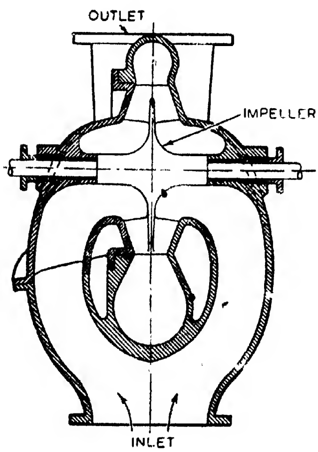


FIG. 37.—SECTION THROUGH A VOLUTE PUMP WITH DOUBLE FLOW

Impellers and Diffuser Rings

The design of the impeller is dependent upon the type of pump, its duty and performance under working conditions. The actual forms differ according to the experience of the various pump manufacturers, and generally are of the open type, as in Fig. 38, or shrouded, as in Fig. 39.

Both impellers and diffuser rings are subjected to severe corrosive and erosive action and are frequently made in Monel or other special alloys. A group of Monel impellers and diffuser rings for turbo-feed, electro-feed, extraction, and circulating pumps will be seen in Fig. 40.

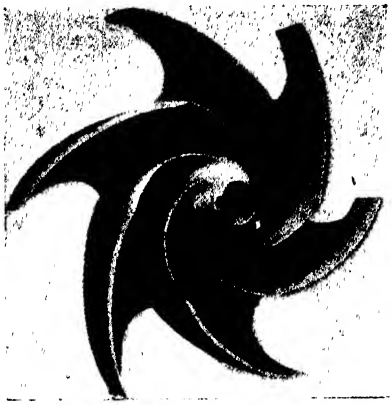


FIG. 38.—OPEN-BLADE IMPELLER

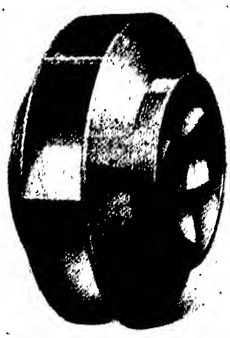


FIG. 39.—SHROUDED-BLADE IMPELLER

Characteristic Curves

The characteristic curves of a centrifugal pump show the relationship of head, power absorbed, efficiency, and corresponding outputs at constant speed. These are obtained from actual tests, and are plotted from readings and measurements taken with the discharge varying in quantity from zero to the maximum output.

Having obtained curves for a given speed, it is possible to construct similar curves over the range of duties to be performed by the pump by applying the laws relating to centrifugal pumps and fans. These are: (1) Volume is directly proportional to the speed; (2) the head is proportional to the square of the speed; (3) the power absorbed is proportional to the cube of the speed.

FIG. 40.—A GROUP OF MONEL IMPELLERS AND DIFFUSER RINGS

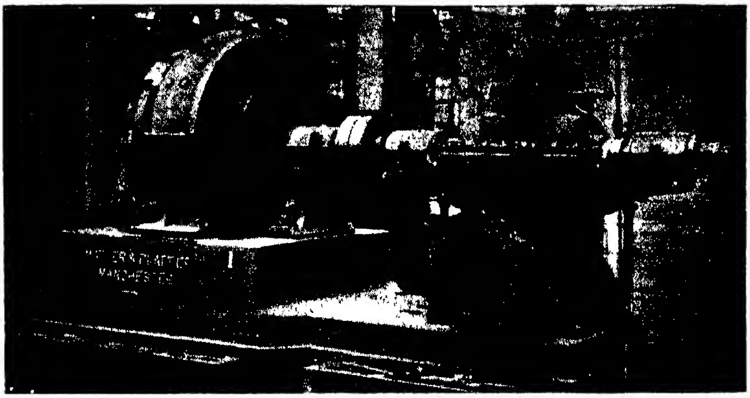


FIG. 41.—SPLIT-CASING GENERAL-PURPOSE PUMP
(*Mather & Platt, Ltd.*)

Split-casing Pump for General Purposes

A split-casing 14–18-in. pump, manufactured by Messrs. Mather & Platt, Ltd., is illustrated in Fig. 41, and a sectional drawing is given in Fig. 43. This type is designed for handling heads up to about 300 ft. The impeller, or two impellers in the case of the two-stage type, is enclosed in a horizontal split casing. This construction affords ready accessibility to the internal parts, which may be removed in their entirety without disturbing the suction and discharge piping joints, a consideration of some importance in large-capacity pumps.

FIG. 42.—MULTI-STAGE TURBINE PUMP
(*Mather & Platt, Ltd.*)

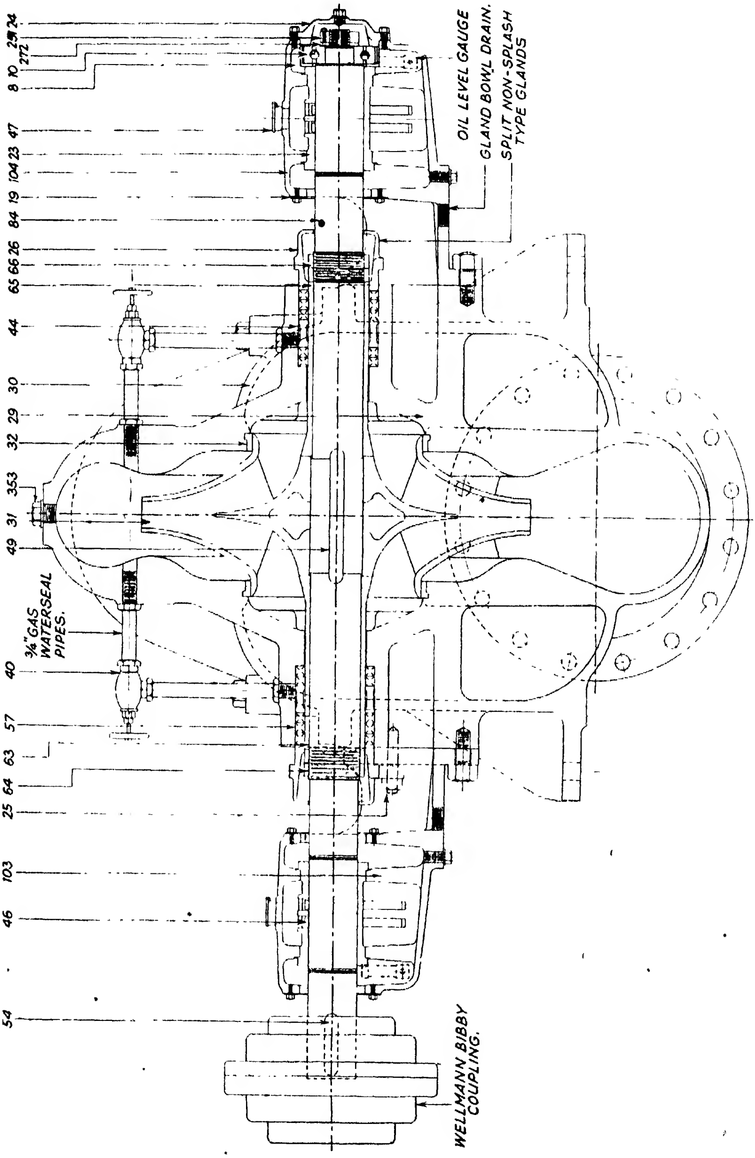


FIG. 43.—SECTION THROUGH GENERAL-PURPOSE PUMP
The list of parts is given on page 158. (*Mather & Platt, Ltd.*)

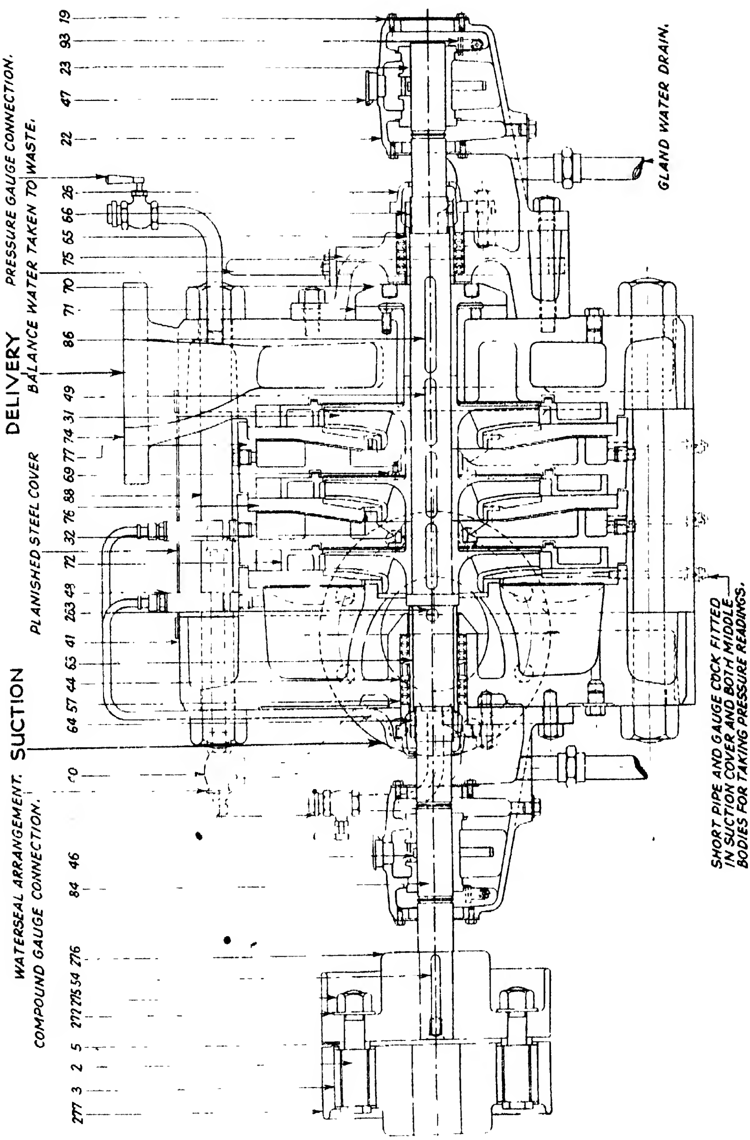


FIG. 44.—SECTION THROUGH MULTI-STAGE TURBINE PUMP

A key to the various parts of the pump is given on page 158. (*Mather & Platt, Ltd.*)

LIST OF PARTS OF GENERAL-PURPOSE PUMP SHOWN IN FIG. 43

Reference No.	*Description*	*Material*	*Reference No.*	*Description*	*Material*
8	Thrust-bearing Cage	T.P.B.	47	Oil Lid (Spring Hinged)	Steel
10	Thrust Bearing	Steel	49	Impeller Key	M.S.
19	Dust Cover	Brs.	54	Coupling Key	M.S.
23	Bearing Bush (Split)	B.B.	57	Packing	Graphite
24	End Cover	C.I.	63	Sleeve (Driving End)	T.P.B.
25	Gland Studs	W.I.	64	Sleeve Nuts (Driving End)	T.P.B.
26	Gland (Split)	Brs.	65	Sleeve (Back End)	T.P.B.
29	Casing (Bottom Half)	C.I.	66	Sleeve Nuts (Back End)	T.P.B.
30	Casing (Top Half)	C.I.	84	Spindle	T.P.B.
31	Impeller	T.P.B.	103	Bearing Housing	C.I.
32	Neck Ring	B.B.	104	Bearing Cap	C.I.
40	Waterseal Valve	G.M.	251	Thrust-bearing Locknut	M.S.
44	Logging Ring	Brs.	272	Locking Ring	M.S.
46	Oil Ring	Brs.	353	Hexagon Plug	W.I.

Abbreviations.—C.I., Cast Iron. M.S., Mild Steel. T.P.B., True Phosphor Bronze. W.I., Wrought Iron. Brs., Brass. B.B., Bearing Bronze. G.M., Gunmetal.

LIST OF PARTS OF MULTI-STAGE TURBINE PUMP SHOWN IN FIG. 44

Reference No.	*Description*	*Material*	*Reference No.*	*Description*	*Material*
2	Coupling Pins	M.S.	65	Back-end Sleeve	T.P.B.
3	Coupling Bushes	Rubber Brs. Lined	66	Back-end Sleeve Nut	T.P.B.
			69	Neck Bush	B.B.
			70	Balance Valve	T.P.B.
5	Distance Sleeve	M.S.	71	Seating and Bush	B.B.
19	Dust Cover	Brs.	72	Guide Tips	T.P.B.
22	Journal Bearing	C.I.	74	Delivery Cover	C.I.
23	Bearing Bush	B.B.	75	Balance-valve Cover	C.I.
26	Gland (Split)	Brs.	76	Partition Plate	C.I.
31	Impeller	T.P.B.	77	Middle Body	C.I.
32	Neck Ring	B.B.	84	Spindle	M.S.
40	Waterseal Valve	G.M.	86	Balance-valve Key	M.S.
41	Suction Cover	C.I.	88	Main Bolts	M.S.
44	Logging Ring	Brs.	93	Oil-level Gauge	Brs.
46	Oil Ring	Brs.	263	Sleeve Key	M.S.
47	Oil Lid	Steel	272	Locking Washers	M.S.
48	Air Cock	G.M.	275	Coupling Nut	M.S.
49	Impeller Key	M.S.	276	Flexible Coupling (Pin Half)	C.I.
54	Coupling Key	M.S.			
57	Packing	Graphite	277	Flexible Coupling (Bush Half)	C.I.
63	Driving-end Sleeve	T.P.B.			
64	Driving-end Sleeve Nut	T.P.B.			

Abbreviations.—C.I., Cast Iron. M.S., Mild Steel. B.B., Bearing Bronze. T.P.B., True Phosphor Bronze. Brs., Brass. G.M., Gunmetal.

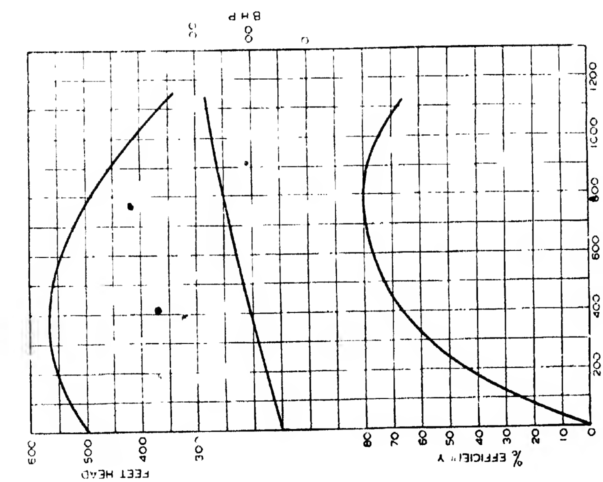


FIG. 45.—CHARACTERISTIC CURVES OF A FOUR-CHAMBER TURBINE PUMP AT 1,400 R.P.M. (*Mather & Platt, Ltd.*)

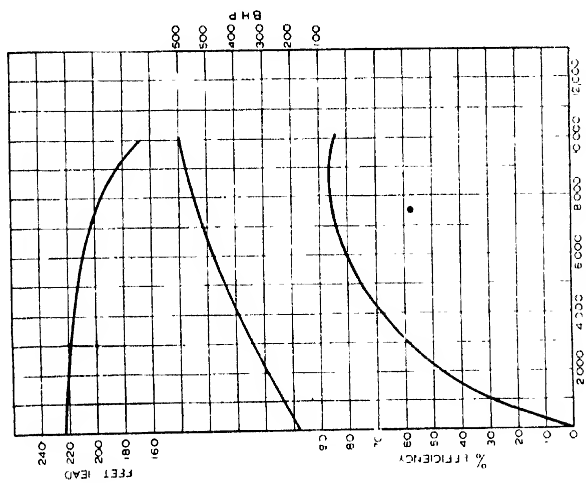


FIG. 46.—CHARACTERISTIC CURVES OF 14–18-IN. PUMP AT 1,000 R.P.M. (*Mather & Platt, Ltd.*)

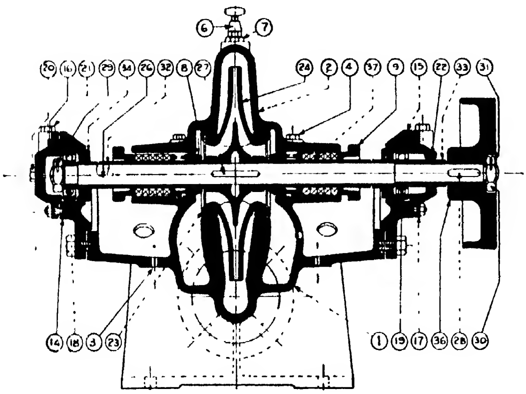


FIG. 47.—SINGLE-STAGE PUMP
(*Pulsometer Engineering Co., Ltd.*)

List of Component Parts.—1. Bottom half-casing. 2. Top half-casing. 3. Gland drain boss. 6. Air cock. 7. Priming plug. 8. Neck bush. 9. Gland. 13. Relief pipe. 14. Ball-bearing housing. 15. Roller-bearing housing. 16. End cover (ball bearing). 17. End cover (roller bearing). 18. Ball-journal bearing. 19. Roller-journal bearing. 20. End plug (ball-bearing cap). 21. Nipples for grease lubrication. 23. Slip rings. 24. Impeller (suction). 25. Impeller (delivery). 26. Spindle. 27. Impeller key. 28. Coupling key. 29. Locknut (spindle). 30. Locknuts (coupling). 32. Sleeves (impeller). 33. Sleeve (coupling). 34. Disc throwers. 35. Centre bush. 36. Half-coupling. 37. Packing.

The impeller is of the double-entry type, which is in hydraulic balance, the shaft being located axially by a ball bearing.

Split glands give easy access to stuffing boxes, and an extended cowl with large drainhole at bottom eliminates water throwing.

The characteristic curves are shown in Fig. 46. The efficiency is well above 80 per cent. at normal output, and power is self-regulating or non-overloading.

Twenty-feet suction lift is obtainable at normal output, and a greater lift when working at a less output.

Multi-stage High-lift Pump

The multi-stage pump illustrated in Fig. 42 is built up in one to ten stages, according to the head. Frequently it is necessary, for considerable heads, to divide the pump into two units working in series and driven in tandem by a prime mover between them.

The middle body, as will be seen from the section given in Fig. 44, is made up of sections of impeller stages, secured together by longitudinal bolts between the end covers. One end cover comprises the suction branch through which the water enters the pump, the head being generated in each successive stage and finally discharged through the delivery cover at the other end.

The impellers are of single-entry type, and the hydraulic end thrust, which is usually very considerable, is taken up by the balance-valve device. They are of bronze or cast iron, turned and bored to gauge. Each impeller is keyed and fitted to the impeller shaft on its own individual key.

Guide Passages

Gradually divergent passages lead from the periphery of each impeller to the eye of the following impeller. The spiral form of these passages ensures

minimum loss and maximum conversion of kinetic energy.

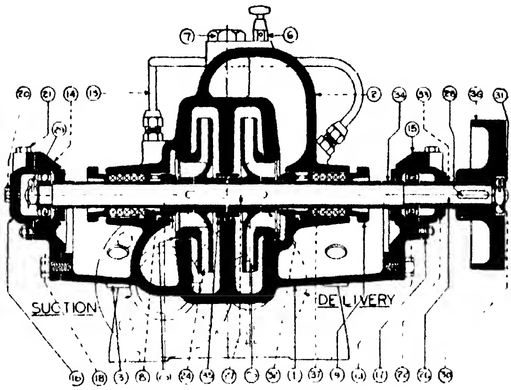


FIG. 48.—TWO-STAGE PUMP
(*Pulsometer Engineering Co., Ltd.*)

Typical characteristics are given in Fig. 45. It will be noted that the head at zero gallons is higher than at normal quantity, thus ensuring the pump will start pumping without difficulty. Also that the efficiency curve has a very wide range, being over 75 per cent. from 550 gallons per minute to over 950 gallons per minute.

This type of pump has a very wide field of application, such as boiler feeding, hydraulic service, colliery drainage, water supply, and the oil industry.

Another type of horizontally split-casing pump by the Pulsometer Engineering Co., Ltd., is shown in section in Fig. 47. This is a single-stage pump discharging into a volute casing.

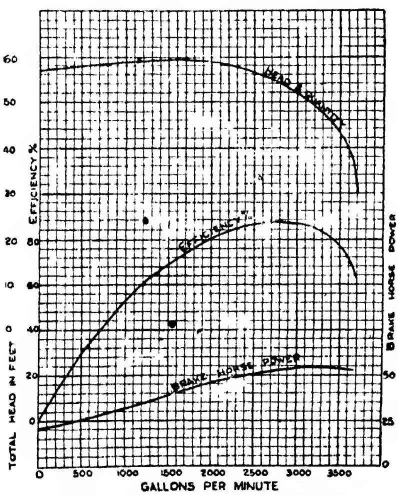


FIG. 49.—CHARACTERISTIC CURVES OF SINGLE-STAGE PUMP

The pump has a double-inlet hydraulically balanced and shrouded impeller with a casing split on the horizontal centre line.

The suction and discharge branches are on the bottom half of the casing, one on each side, and pointing in a horizontal direction.

The pump can be arranged right or left handed by turning the casing with impeller round and reversing the shaft.

The shaft is supported by a roller bearing at the coupling end and a double-purpose ball bearing at the opposite end. The bearings are held axially by covers in solid housings, and the housings have turned flanges with spigots which

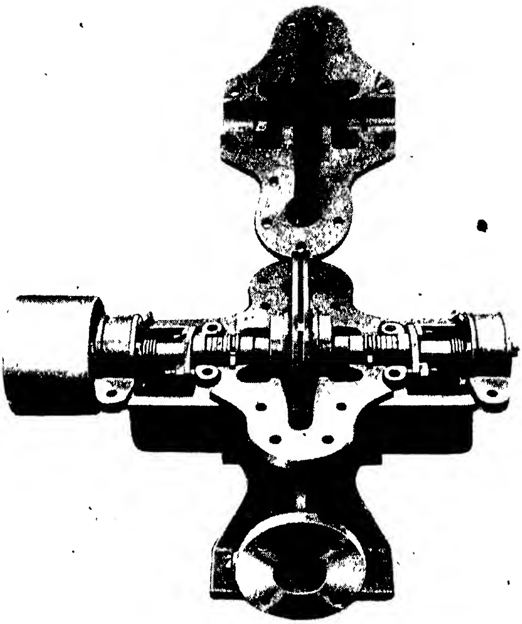


FIG. 50.—SPLIT CASING PUMP WITH COVER REMOVED
(*Ruston & Hornsby, Ltd.*)

fit into recesses in the pump casing to ensure automatic alignment.

Two-stage Pump

The two-stage pump shown in section in Fig. 48 is similar in general construction to the single-stage pump, but has two impellers of the single-inlet type, made right and left handed and mounted on the shaft back to back. By this arrangement the hydraulic thrust is balanced and slip rings are only fitted on the inlet side of the impellers. The discharge from the first impeller is carried through a passage on the top of the pump to the eye of the second impeller. Lantern rings are arranged in the neck bushes, connected together by an upper pipe, thus providing a pressure seal

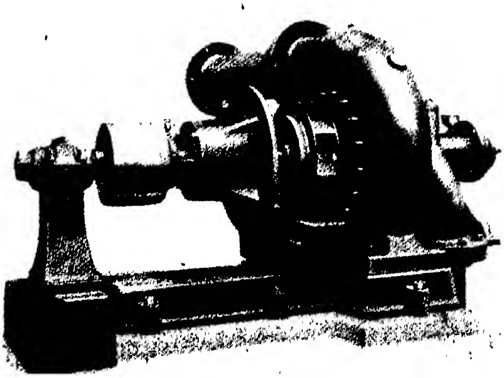


FIG. 51.—METHOD OF REMOVING IMPELLER FOR CLEANING PURPOSES
(*Ruston & Hornsby, Ltd.*)

on the first-stage stuffing box and relieving the pressure behind the stuffing box of the second stage. This type of pump covers heads up to 500 ft.

Typical characteristic curves are shown in Fig. 49.

The pump in Fig. 50 is intended for heads up to 180 ft. The impeller is of the double-inlet shrouded type and is normally made of cast iron. End movement is prevented by the stuffing-box bushes. The various components are clearly shown in the illustration.

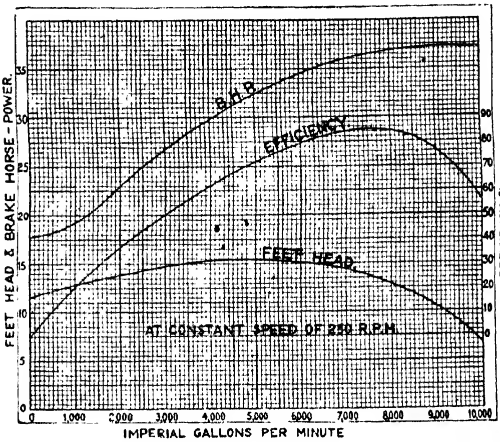


FIG. 52.—CHARACTERISTIC CURVES FOR TWO-STAGE PUMP

Another type, also by Messrs. Ruston & Hornsby, Ltd., will be seen in Fig. 51. This is designed for heads up to 35 ft., and is provided with a removable cover, by means of which the impeller can be withdrawn, inspected, and cleaned.

Characteristic curves are shown in Fig. 52. A high efficiency is obtained over a wide range in output.

Condensate Pumps

Condensate pumps are built for high-vacuum service handling condensate, chilled water, and similar low net positive suction-head applications. They can be designed for either industrial or marine uses.

Among the principal features of these pumps may be mentioned the following: impellers with large suction areas; extra-deep stuffing boxes under discharge pressure and arranged for external seal; shaft sleeves packed to prevent air leakage and oil-lubricated ball bearings. The pumping units are manufactured for capacities up to 4,000 gallons per minute and for heads to 600 ft. or more.

The horizontal pump shown in Fig. 53 is manufactured by Ingersoll-Rand Co., in four standard sizes of capacities up to 1,000 gallons per minute and heads up to 230 ft. A larger range of horizontal pumps is available for capacities of 4,000 gallons per minute and heads up to 575 ft. These pumping units are also suitable for heater-drain service, where condensate temperatures are 200° F. or below.

Vertical condensate pumps are particularly suitable for applications where limited available net positive suction head would necessitate extremely large,

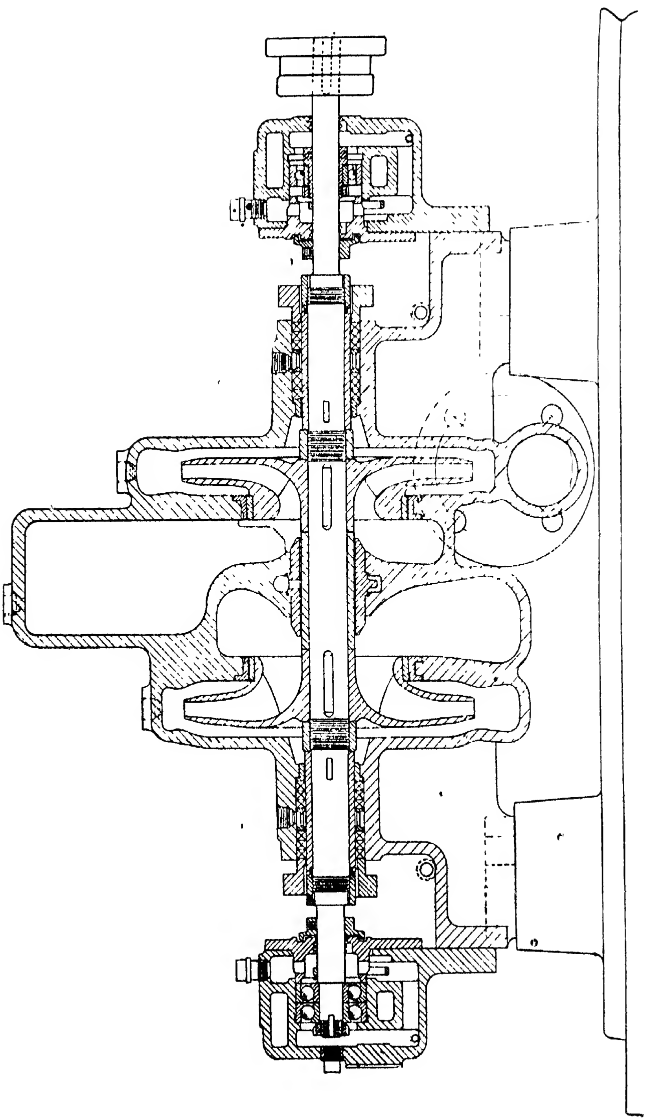


FIG. 53.—CROSS SECTION OF A CAMERON HORIZONTAL CONDENSATE PUMP
(*Ingersoll-Rand Co.*)

low-speed horizontal pumps. They are also ideal in situations where floor space is at a premium. These pumps are of the vertical turbine type, with mix-flow impellers. The pumping element is enclosed in a shell which forms the suction well. This shell is installed below the floor line. Available suction head is, therefore, the distance between the water level in the condenser hotwell and the first-stage impeller which is at the bottom of the pump. The pump shell and discharge column can be furnished in a length suitable for the net positive suction-head requirements. There is only one stuffing box which is under discharge pressure. The suction nozzle is located at the top of the unit, but it may be provided on the barrel below the floor mounting plate if underground suction piping is required. Because of the vertical construction these units are self-venting.

The type of pump described above is manufactured by Ingersoll-Rand Co., for capacities to 1,500 gallons per minute and heads to 575 ft.

CENTRIFUGAL WELL AND BOREHOLE PUMPS

The development of centrifugal pumps for wells and boreholes has been very remarkable during the past fifteen years, and these units have proved most reliable under the exacting service demanded by waterworks and similar undertakings.

It is interesting to examine the circumstances which are responsible for this development. Until the introduction of the centrifugal pump, most waterworks obtaining their supplies from deep water bearing strata relied upon plunger pumps installed in wells and boreholes, and generally driven by steam engines. These units were highly efficient and dependable. Some, indeed, have been in service for nearly a hundred years.

Most engineers are familiar with steam-engine-driven pumps, and their workmanship especially is remarkable when it is considered that at the time of their manufacture high-precision machine tools were not available, and a good deal of the work spent on them was hand labour. Unfortunately, in spite of their high efficiency, the cost of installation, both from the point of view of the machinery itself and buildings, is to-day prohibitive, and the centrifugal pump has the great advantage that for a given size of borehole or well it can discharge greater quantities of water than the plunger type of pump.

The centrifugal pump is able to accommodate itself in the fullest degree to operation by modern high-speed prime movers, thus assuring relatively low cost of the machinery, and permitting small buildings to be used with light foundations.

It is true that although the efficiency of the centrifugal pump has substantially improved during recent years, the plunger pump still remains the more efficient of the two types. Modern practice, however, assigns a life to machinery which is shorter than that contemplated by the pioneer manufacturers of steam reciprocating pumps, so that from an economic point of view the effect of lower power costs when considered in conjunction with capital charges is not overwhelmingly less in favour of the plunger pump.

Advantages of the Centrifugal Pump

The centrifugal pump has the advantage that maintenance charges are extremely low, since it is seldom necessary to withdraw a centrifugal pump for repair unless the water is very sandy, and many centrifugal well pumps and borehole pumps have been running from ten years to fifteen years without repair and without showing marked loss of efficiency.

Type Adopted

As far as the actual pump is concerned, the type used for deep well and borehole installations presents no special features of design, most manufacturers utilising multi-stage turbine pumps or multi-stage single-suction non-diffuser pumps modified for vertical operation.

Power Transmission

Two systems of power transmission are in use: one in which the pump is driven by means of a transmission shaft operated by a driving head on the surface, and the other in which the transmission shaft is eliminated and a submersible motor, built up as a unit with the pump, is employed. In the former system the pump is suspended at the desired depth in the well or borehole by a steel rising main made up in short lengths, and the pumpshaft is connected to the prime mover by an intermediate transmission shaft, which is housed in the rising main itself, the bearings being supported by sandwich plates bolted between each length of rising main.

The suspension main, with its internal shafting, is bolted to a substantial wellhead baseplate, carried on girders or sometimes on concrete foundations, this baseplate forming a mounting for the pedestal which supports the prime mover, or, in the case of engine-driven plant, the gear. To the suction flange of the pump is bolted the suction pipe and strainer, and in some cases a foot valve.

Foot Valve

Usually in boreholes and wells, when pumping ceases, the water level rises, sometimes considerably above the pumping level which obtains when the plant is delivering at its rated capacity. In these circumstances it is not necessary to fit a foot valve, as when the pump is started it is primed and will pick up water instantly. Sometimes, however, it is necessary to start the pump with the water level below the pump itself, and for these conditions a foot valve is necessary.

Another reason for omitting the foot valve is that with motor-driven plant, the run-back action caused by the reversal of flow in the rising main of the pump when the power is switched off is of value in the case of water containing sand, as it washes out any sand that may have accumulated in the water passages of the impellers, guide rings, and clearance spaces; whilst with a foot valve the column is static when the pump is stopped, and there may be some settlement of sand, which is liable to cause wear.

Special Considerations

The following points need careful attention in preparing designs for centrifugal well and borehole pumps:

(*a*) The rising mains are usually made up of welded steel pipes with welded steel flanges, and special precautions are necessary to ensure that the welds are perfectly sound. A rising main fracture might cause the pump assembly to drop in the well or borehole, which might prove a costly matter to rectify.

(*b*) The transmission shaft between the pumpshaft and the prime mover must be proportioned so that there is no possibility of the shaft running on a critical speed. Special care in this respect is needed when the prime mover is a variable-speed machine. Some pumps are designed to run through the first critical speed, but this is not considered good practice, although it may be necessary on occasions.

(*c*) The bearing material for the transmission-shaft intermediate bearings must be carefully chosen. Usually bearings are made of lignum vitæ, although rubber bearings have been used with great success, and some manufacturers even employ special bronzes.

In each case water is relied upon as the lubricating medium, and where the water is sandy, special precautions must be taken to maintain a filtered water supply to the bearings, except in the case of rubber bearings, which are not considered to be affected by sand.

When lignum vitæ is used, the bearings, together with the shafting, are usually enclosed in a steel-shaft tunnel to which filtered water is supplied at a pressure in excess of that in the external main, so that if the shaft-tunnel joints leak, there will be a leakage of clean water into the main, and not a leakage of impure water from the main into the shaft tunnel.

Combined Borehole Pump and Force Pump

When a combination of borehole pump and force pump are mounted in the same assembly, a tapping is taken from the force-pump discharge, the water so obtained being passed through a filter of the duplex type before its introduction into the shaft-tunnel compartment.

Where the external head is insufficient to boost up lubricating water to the necessary pressure a separate pump is employed. This may be incorporated in the main pump assembly or, as is more usual, a separate motor-driven pump is supplied.

(*d*) Couplings must be designed so that they are readily assembled or dismantled without damage to the components. To give the smallest possible diameter, muff couplings are employed. The design of these varies with the manufacturer, but all have devoted a great deal of thought to the subject.

(*e*) The automatic balancing valve associated with the turbine pumps is very rarely used for well or borehole pumps, since it is liable to wear and may need replacement if the water is sandy, and this could only be done by withdrawing the pump from the well or borehole. Usually the thrust is taken by a thrust bearing fixed in a pedestal, which is bolted to the baseplate at the wellhead.

For small pumps it is sometimes possible to use with success ball bearings, but for larger pumps, where the hydraulic end thrust of the single-suction impellers is considerable and the weight of moving parts to be sustained is high, a " Michell " thrust bearing is usually provided.

Some manufacturers use a hydraulically balanced pump and provide a light thrust bearing to take the weight of the moving parts.

(*f*) When pumping plant has to be installed in deep boreholes, it is important that careful calculations should be made to gauge accurately the stretch of the rising main and the pumpshaft with its transmission shaft, so that the impellers may be correctly located in the casing. Incorrect calculations and setting of the impellers may cause considerable damage, for the impellers may sit on the bottom of the casing, in which case the transmission shaft is in compression instead of tension, or, vice versa, they may be pulled up too high, resulting in excessive thrust.

It is important that the method of setting the impellers should be one which gives an accurate adjustment with close control.

The Head against Pump

In those cases where the external head is considerable, it is economical to provide a number of stages in the borehole or well sufficient to deliver the water to the engine-room floor level, introducing just below floor level on the same transmission shaft a turbine pump of sufficient stages to deal with the external head. This force may be inverted so as to partially balance the hydraulic thrust of the well or borehole pump, thus relieving the load on the thrust bearing considerably. It must be appreciated that the power absorbed by the thrust bearing is not inconsiderable, and it varies almost directly as the speed.

Under some conditions of operation a combined assembly is not practical from the point of view of service conditions, and in such cases it is usual to provide a separately driven force pump.

Loss of Power in Shafting

The loss of power in shafting, couplings, bearing, and thrust bearings is important, especially in the case of pumps where the water horse-power is small. Under such conditions the power losses of the shaft, etc., may be a high proportion of the total power demanded of the prime mover. In fact, it is the ratio of useful work to total power which determines the limits of usefulness of the centrifugal type of pump for these installations.

Nevertheless, centrifugal pumps are frequently installed in small boreholes when transmission thrust losses are high, on account of their reliability and the fact that they will work for much longer periods than plunger pumps without attention when the water is sandy.

Method of Driving

Pumps may be driven by vertical direct-current or alternating-current motors, in which case they are directly coupled through a flexible coupling to

the transmission shaft, the motor being bolted to the thrust pedestal; or they may be driven by steam or diesel engines, when gears are employed to secure the right-angle drive and to allow suitable adjustment between the engine and the pump speed.

Vertical steam turbines have been used with great success as prime movers. They are the geared type, being mounted directly above the transmission shaft and flexibly coupled to it, a similar assembly to the motor-driven plant. Vertical condensers have been developed which snug up to the turbine and result in a great saving of floor space.

Electric Drive

Constant-torque machines, such as the electric motor and turbine, introduce no difficulties with regard to resonance and torsional vibrations, but where reciprocating engines are employed, careful calculations have to be made to ensure that trouble due to overrunning the speed of the unit will not arise.

Diesel Drive

Where diesel engines are employed, the use of the multi-cylinder engine giving the smallest possible cyclic irregularity is recommended, and care must be used in the choice of the flexible coupling between the shaft and the slow-speed shaft of the gearbox, since a well-designed coupling is able to provide a substantial damping effect.

System for Several Boreholes

In some installations the boreholes from which the water is drawn are scattered over a comparatively large area, and under these conditions it is usual to adopt a diesel electric drive. A central building houses the generating plant, and small pump houses are erected over the boreholes, in which are installed motor-driven plant.

The power is transmitted from the generating station to each of the boreholes, and the switch gear is arranged in the engine room, and is of such a type as will allow remote starting, stopping, and speed control from the generating station.

Centrifugal deep-well and borehole pumps will run continuously with no more attention than an occasional examination of the oil level of the thrust bearing, and for electrical machinery methods are available to shut down the plant automatically in case of failure of one of the essential functions on which their running depends.

Automatic Arrangements for Emergencies

The automatic features usually provide for the following emergencies:

(*a*) Failure of water supply in well or borehole.
(*b*) Failure of lubricating water supply to transmission shaft bearings.
(*c*) Excessive temperature of thrust-bearing oil.
(*d*) Excessive temperature of motor bearings.

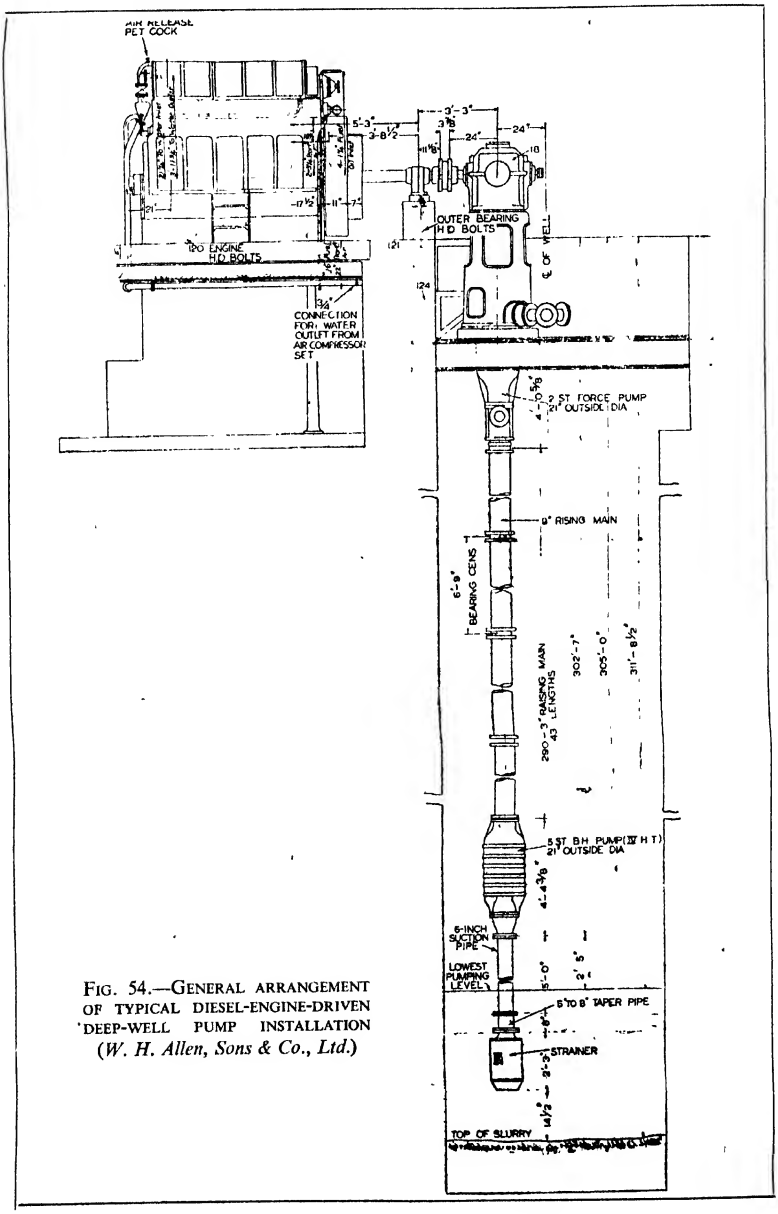


FIG. 54.—GENERAL ARRANGEMENT OF TYPICAL DIESEL-ENGINE-DRIVEN DEEP-WELL PUMP INSTALLATION (*W. H. Allen, Sons & Co., Ltd.*)

The plant may be shut down if there is a falling off in the yield of water, resulting in the water level falling below the pump, by a simple electric device operated by the water-level indicator, contacts being connected electrically to an interlock on the motor starter.

Failure of the transmission-shaft bearing lubricating supply, whether it be on the inlet side to the filters or on the outlet side from the filters, might damage the bearings, and to avoid this a pressure switch is provided which is interlocked with the motor starter, and will cut off the current supply to the motor if the pressure rises unduly, owing to clogging of the filters when the water supply becomes restricted, or if it drops, due to obstruction or fractured pipe on the inlet supply to the filters.

To prevent damage to the plant from excessive temperature of the thrust bearing or motor bearings, thermostats are provided interlocked with the motor starter and set to shut down the motor if a predetermined temperature is exceeded.

The dependability of centrifugal well and borehole pumps will be understood when it is considered that pumps are often installed with shafts exceeding 300 ft., and are capable of running continuously day and night for years without serious wear or loss of efficiency with a speed of over 1,400 revolutions per minute.

These pumps have been installed in large numbers, not only in waterworks, but in factories, plantations, and mines. In the last named they are often installed in a disused pit shaft, which is used as a collecting sump for the active workings of the mine, and saves great expense, since all the working machinery is above ground level and the cost of sending down regular shifts for attending the pumps is avoided. In one case a pump of this type was operated automatically, being inspected once a week only.

A general arrangement and sectional arrangement of a typical diesel-engine-driven well-pump installation are illustrated in Figs. 54 and 55. The plant is designed to deliver 25,000 gallons of water per hour from a depth of 305 ft. below engine-room floor level, against an external static head of 127 ft., a total head including friction of 437 ft.

Five stages of the pump are suspended from just over 290 ft. of rising main from the wellhead, and just below delivery head a two-stage force pump is fitted. The suspension main is 9 in. internal diameter, the bearings being spaced 6-ft. 9-in. centres.

The pumps run at a speed of 1,200 r.p.m., and the power required to drive them, including gear and other losses, is 84 b.h.p. The drive is transmitted from the engine shaft to the pump transmission shaft by a totally enclosed spiral bevel gear.

The prime mover is a five-cylinder engine having a continuous rating of 110 b.h.p. at 350 r.p.m., and the fuel consumption per w.h.p. works out on average running conditions to 0·57 lb.

The pumps are of the turbine type fitted with guide rings, each stage building up a definite increment of head of 62·5 ft.

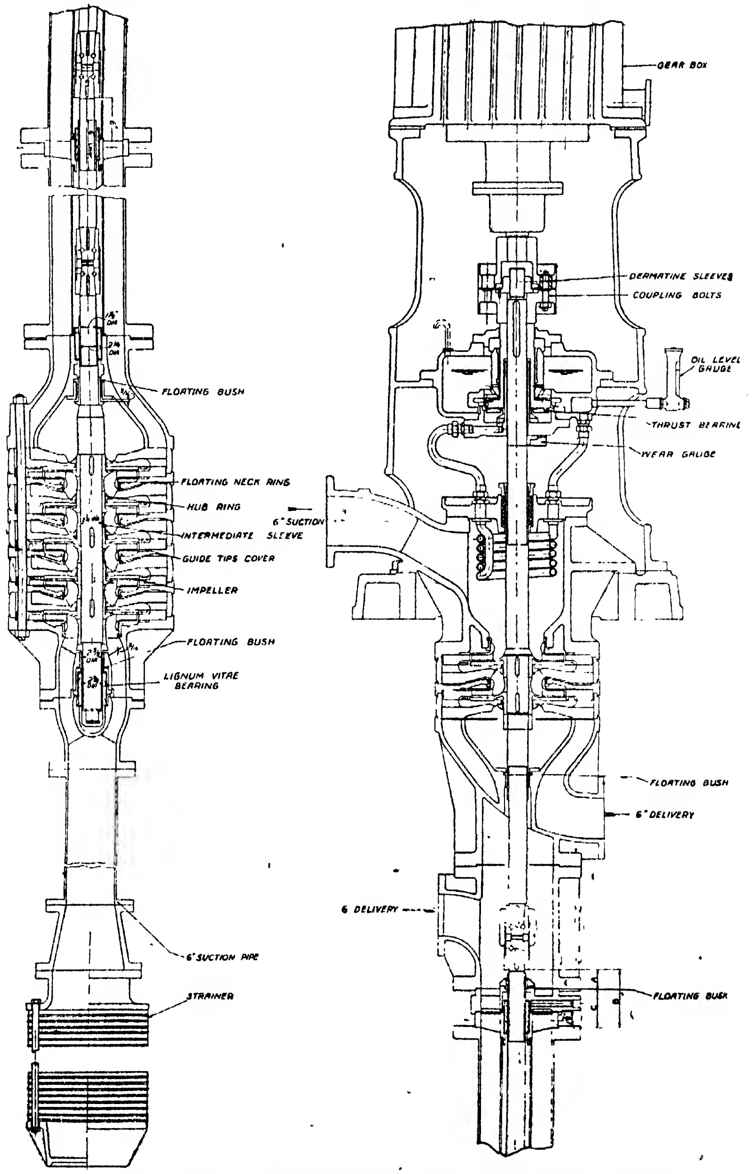


FIG. 55.—SECTIONAL ARRANGEMENT OF BOREHOLE AND FORCE PUMPS
(*W. H. Allen, Sons & Co., Ltd.*)

Thrust

The hydraulic thrust of the force pump is opposed to that of the borehole pump, so that when the pump is running up to speed the Michell thrust bearing situated in the wellhead pedestal is relieved of practically all its load.

The engine is started by compressed air, an air receiver of sufficient capacity to give four to five consecutive starts being provided, together with an engine-driven compressor for recharging. The compressed-air system operates on a pressure of 300 lb. per square inch.

Fuel Tanks

A day-service fuel tank is mounted above the engine, fuel falling by gravity into a Duplex filter, from which it is distributed to the fuel pumps. The tank holds a sufficient supply of fuel oil to enable runs of twelve hours to be made at full load without replenishing. A 10-ton fuel tank gives a bulk supply of oil, transfer from the main storage tank to the day service tank being made by a semi-rotary hand-operated fuel-transfer pump.

Engine-jacket Water

The engine-jacket water is obtained from an overhead tank, fed from a low-pressure main under ballcock control, the hot water being returned to an adjacent reservoir. Modern practice is to maintain a comparatively high inlet-cooling water temperature, and to give some control in this respect a preheater is fitted, heat being taken from the exhaust system.

The plant illustrated was manufactured by Messrs. W. H. Allen, Sons & Co., Ltd., and is installed in a waterworks near London. It is extremely economical in running and has proved absolutely dependable.

Submersible Borehole Pumps

One of the latest developments in borehole pumping machinery, as applied to boreholes and mineshafts, is a pump in which the vertical driving spindle is eliminated, the motor being made submersible and built up as one unit with the pump. This wet motor pump is designed to raise water from great depths in the simplest and most efficient way. The installation is simple, inexpensive to install, maintain, and run.

No pumphouse is needed to protect the submersible pumping unit. The headpiece from which the rising main pipe and unit are suspended can be placed in a pit immediately under the surface, and floored over with a manhole cover as chequer plating, so that the floor space above the pumping unit can be put to normal use.

The general arrangement of a Hayward-Tyler "Electromersible" pump is shown in Fig. 57, and the sectional arrangement in Fig. 56 reveals the details of its design.

The centrifugal pump portion of the unit consists of multiple stages, each comprising an impeller (of the radial flow type), a guide vane, and a return guide ring. The impellers are mounted on a stainless-steel pump shaft with

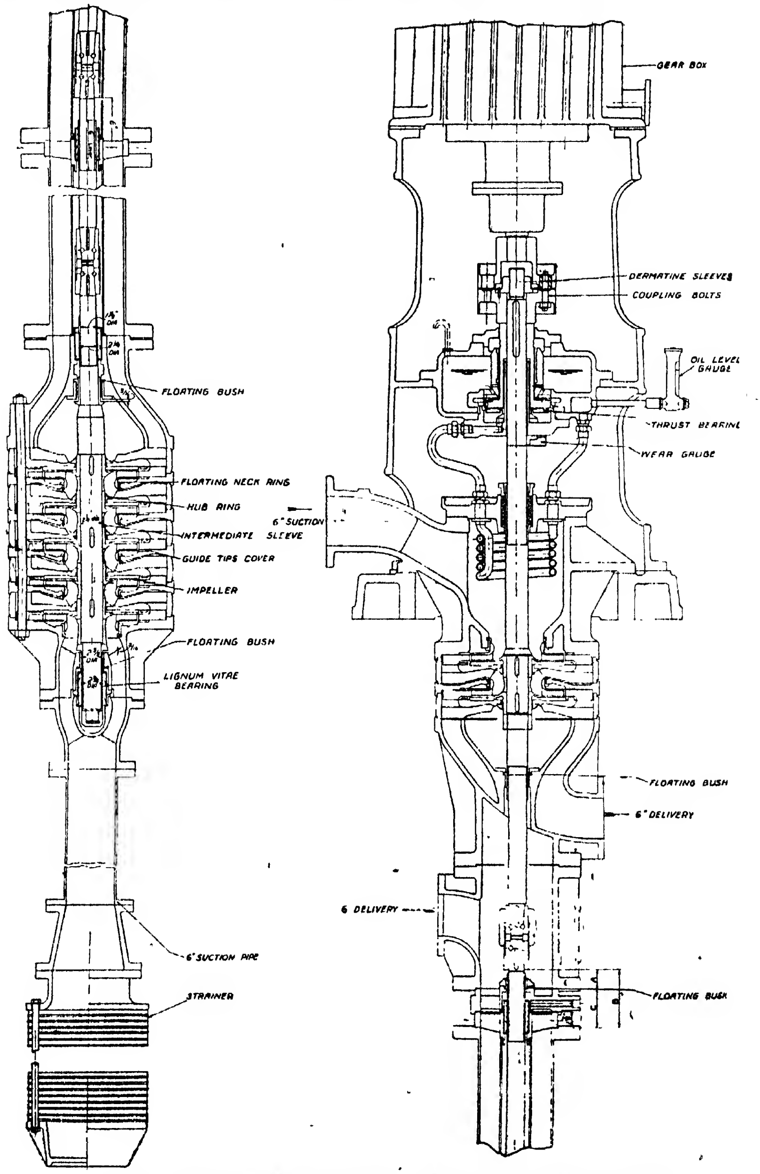


FIG. 55.—SECTIONAL ARRANGEMENT OF BOREHOLE AND FORCE PUMPS
(*W. H. Allen, Sons & Co., Ltd.*)

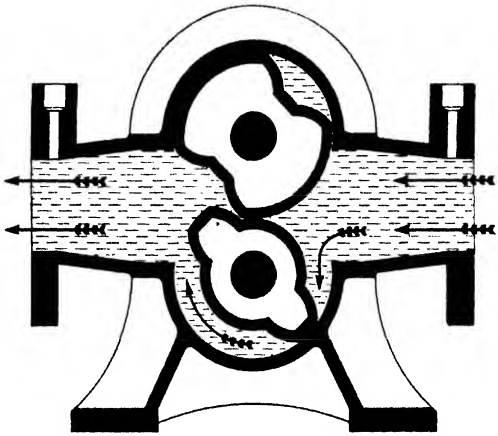


FIG. 58.—A DRUM PUMP
(*Drum Engineering Co., Ltd.*)

A high-duty thrust bearing of the water-lubricated tilting-pad type, similar to the Michell bearing and having a self-aligning seat, is located at the foot of the motor. This bearing is capable of taking the pump thrust as well as the weight of the rotating assembly, and thus makes hydraulic balancing of the pump unnecessary.

The current supplied to the motor is carried to it by a 3-core C.T.S. cable supported by the rising main and clipped to it by means of adjustable clips. The rubber cable is readily attached to and detached from the motor leads by means of three single-pole waterproof cable couplings which embody copper split connectors.

To limit the starting current in compliance with local requirements, auto-transformer starters are normally used with wet motors of 4 h.p. or more. These starters, in addition to preventing high supply-line starting currents, usually incorporate a number of motor protection devices. It is usual to fit an ammeter, as this will give a good indication of how the motor is operating.

The switchgear may be either of manual or automatic type, and, with the latter, variations in the level of the water in the reservoir or borehole can control the starting and stopping; the installation will thus require no supervision, and the motor will never operate in a dry borehole. Alternatively or additionally, time clock, pressure control, or remote push buttons can be incorporated in the control circuit.

The "Electromersible" pump is made by Messrs. Hayward-Tyler & Co., Ltd., in some fifteen standard sizes, suitable for boreholes over 8 in. inside diameter.

ROTARY PUMPS

The term "rotary pump" is usually applied to a pump in which a rotor actuates pistons, plungers, drums, or gears, and displaces liquids by positive action. They are most compact and usually without valves, and are mainly used for pumping small quantities of liquids against large heads or comparatively large quantities of liquids against low heads. Rotary pumps are constructed in many designs, and are utilised for such work as circulating water

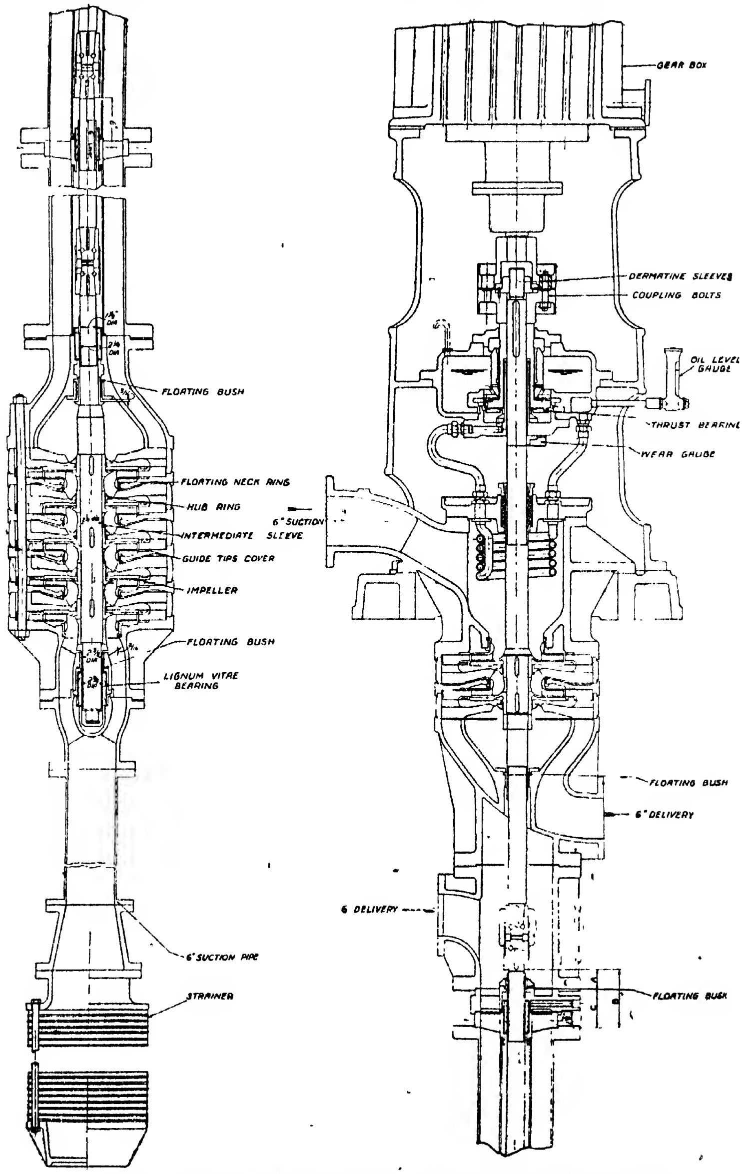


Fig. 55.—Sectional arrangement of borehole and force pumps
(*W. H. Allen, Sons & Co., Ltd.*)

high suction capacity, while the enclosure of the rotors is such that motion of the fluid takes place largely in an axial direction, hence the actual velocity of inlet or expulsion barely succeeds the peripheral velocity of the rotors. These pumps are made in eight sizes and operate at speeds of 100 r.p.m. to 1,500 r.p.m. with a normal discharge of 0·53–1,500 gallons per minute and a range of pressures up to 300 lb. per square inch.

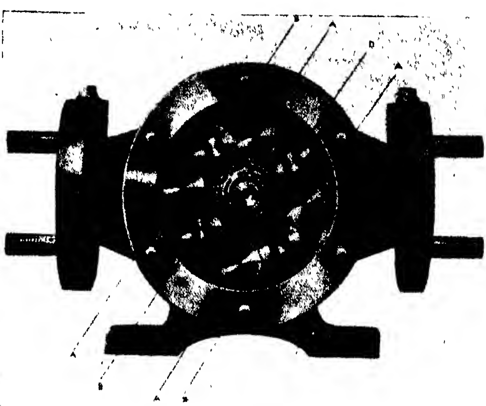


FIG. 62.—ROLLER PUMP
(*Drum Engineering Co., Ltd.*)

The actual discharge of a pump depends upon the volumetric efficiency obtained at a given pressure and speed of operation, and is influenced also by the viscosity of the liquid. The curve given in Fig. 61, showing maximum pitch-line velocities for viscosities up to 10,000 seconds Redwood, should therefore be taken into account when determining the size and speed of pump for a given duty.

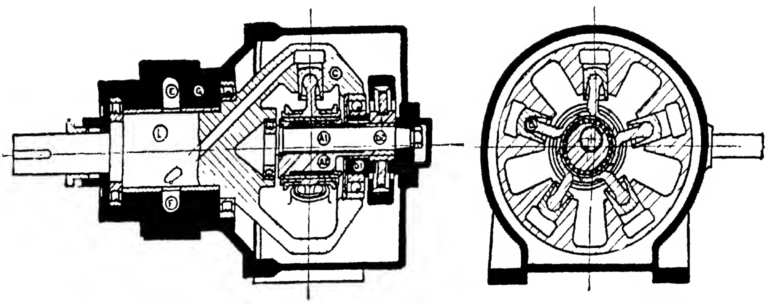
Another type of pump is designed to deal with large quantities of fluid at relatively low pressures, being particularly suitable for use in the lubrication systems of gear units and for machine-tool lubrication and coolant supply. There are two sizes, with two arrangements of the smaller size, all of them designed to operate against a maximum pressure of 25 lb. per square inch. Rate of discharge ranges from 0·6 to 2 gallons per 100 revolutions, and speeds from 300 to 1,200 r.p.m. Fig. 60 illustrates this particular type.

Roller Pump

A positive displacement roller pump is shown in Fig. 62. Two sets of rollers are used, which are marked *A* and *B*. The *B* rollers are termed "pushing rollers," and are fixed to the rotor and turn with it; they push the rollers in between them which are loose. When the pump is revolving, the loose rollers fly out to the periphery of the pump and, due to eccentricity of the rotor centre to the centre of the casing, adjacent rollers approach and recede from each other. This produces the pumping action, and by arranging suitable ports, the liquid is drawn into and discharged from the pump.

Rotary Pump with Automatic Stroke Control

The Hele-Shaw Beacham rotary pump with automatic stroke control is used in conjunction with a ram or piston for operating presses, lifts, cranes, and



Longitudinal section of pump. Section of pump, end elevation.

FIG. 63.—ROTARY PUMP WITH AUTOMATIC STROKE CONTROL
(Greenwood & Batley, Ltd.)

machine tools, and in conjunction with hydraulic motors for operating hydraulic variable-speed gears. The normal running speeds are high, and the pump can be directly coupled to an electric motor without the intervention of any form of gearing. The high speeds and the use of three or five cylinders of small capacity give an even flow on the delivery side of the pump. This type is being used in extruding operations under pressure, when detectable pulsations would spoil the finished work, and also in the operation of testing machines, the pump being directly connected to the hydraulic cylinder containing the ram and eliminating any form of hydraulic accumulator.

An important feature is the variable delivery from zero to the maximum capacity of the pump, and the reversal of flow without stopping the pump or changing its direction of rotation. The variable delivery may be effected by hand control or may be governed by the pressure in the pipeline or container into which the pump is discharging. The automatic stroke control is also important, in that it allows the pump to discharge at long periods, without attention, into pressure systems, with the knowledge that the pump is only using sufficient power to maintain the required pressure without exhausting against a safety valve.

The pump is shown in Fig. 63, and consists of a cylinder block *C*, having a number of radial cylinders revolving within a cylinder casing to which is secured the crank A^1 A^2. The source of power is coupled to the cylinder-block extension which forms the shaft and central revolving valve *L*. The throw of the crank is variable, from a maximum at one side of the centre of rotation of the cylinder block to a similar maximum at the opposite side of the centre, the crank passing through a position of "no throw" at the actual centre of rotation. The pistons are connected to the crank by the connecting rods *B*.

The crank consists of two eccentrics A^1 A^2, which fit one within the other, the throw of each being equal. By turning the eccentrics relatively to each other, so that the respective "throws" cancel out, the outer circumference of the pair becomes concentric with the axis of rotation of the cylinder block, and no

movement of the pistons within the cylinders occurs when the cylinder block is revolving. If, from this position, the eccentrics are rotated about each other in either direction, the pistons will move up and down the cylinders as the cylinder block revolves, and the capacity of the pump increases in proportion to the amount that the crank is offset.

The pressures obtainable are as high as 2,240 lb. per square inch in the smallest standard size, and 1,500 lb. for intermittent or 750 lb. for continuous working in the largest.

THE MERRYWEATHER TURBINE FIRE PUMP

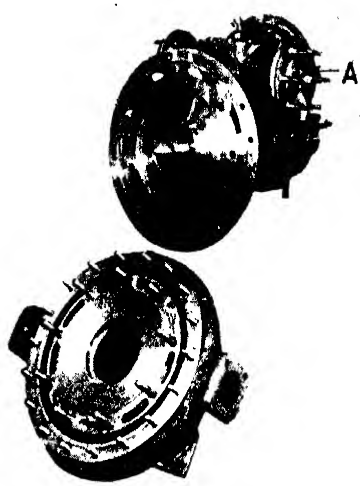


FIG. 64.—MERRYWEATHER TURBINE FIRE PUMP

The casing is removed from the main pump in order to show the shaft and impeller with the encircling diffuser outlets through which the water is taken to the delivery outlets

The transverse mounted reciprocating pump is clearly seen at (*A*).

The pump shown in Fig. 64 has special characteristics to provide the wide range of quantities and pressure required by the modern fire brigade.

The output obtainable from the pump is, of course, dependent to a large extent on the power output of the engine driving it, but it is designed to be capable of delivering 600 gallons per minute at 180 lb. per square inch; 875 gallons per minute at 140 lb. per square inch, and 1,100 gallons per minute at lower pressures.

The pump is of the single-stage type, with casing and impeller of gunmetal. The shaft is of high tensile steel protected at the gland by a stainless-steel sleeve. The impeller, which is carefully balanced, is overhung, so that no bearing is necessary in the suction waterway.

The impeller shaft is supported in ball and roller bearings spaced well apart so as to ensure the absence of vibration of the impeller at all speeds of rotation. A stuffing box is provided on the delivery side only of the impeller, with a specially designed centrally loaded hand-adjusting mechanism, which ensures easy and even adjustment and prevents overtightening of the gland.

Delivery Outlets

The delivery outlets are two in number, and each is provided with branches and valves for controlling two or three lines of hose.

The diffuser vanes, the outlets of which are visible in Fig. 64, are formed in the main pump casting, and discharge into the annular spaces in the casing

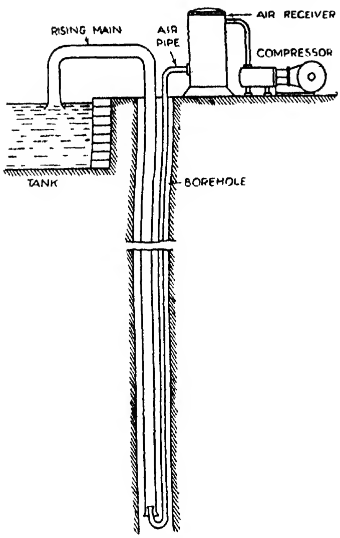


FIG. 65.—ARRANGEMENT OF COMPRESSED-AIR PUMPING PLANT

visible in the lower picture, which connect with the pump deliveries.

The suction inlet, screwed for 6-in. hose, is centrally situated at the rear of the casing.

Priming

The priming of the pump is carried out by a reciprocating vacuum pump (see *A*) operated by a toothed-wheel gearing from the main pump spindle. The gearing is put in and out of action by a hand-operated coned clutch. The whole of the vacuum-pump driving gear is housed in an oiltight gunmetal casing secured to the front cover of the main pump. Water can be lifted from a depth of 24 ft. in 20 seconds.

The vacuum pump is of the horizontally opposed single-acting type, the plungers being operated by a two-throw crank. The valves and seatings are arranged in the two end covers of the cylinder, the delivery valves being of so large an area relative to the suction valves as to facilitate to the greatest possible degree the egress of any water to the pump barrel.

Furthermore, the end covers are spring loaded in a special manner to permit them to yield under excessive water pressure, so that the covers themselves act as efficient relief valves for the cylinder barrel.

COMPRESSED-AIR PUMPING

The system adopted for raising water and other liquids by compressed air is comparatively simple, and consists of forcing air into the open end of a rising main deeply submerged in the liquid. The air impregnates the liquid and reduces its density, with the result that the weight of a column of liquid and air inside the rising main is less than the weight of an equal column of liquid surrounding the main. When in action, rising air bubbles and layers of air, together with the head of liquid acting on the inlet of the rising main, forces the liquid to rise a distance dependent upon the amount of submergence and the pressure of the air.

The advantages of this system of pumping are: simplicity; borehole need not be perfectly straight or vertical; sand or other impurities are not detrimental to the plant, and hot liquids can be raised as easily as cold.

The main disadvantages are: low efficiency, the maximum being about 50 per cent. and the average about 30 per cent., and the borehole must be deep enough to obtain the necessary submergence.

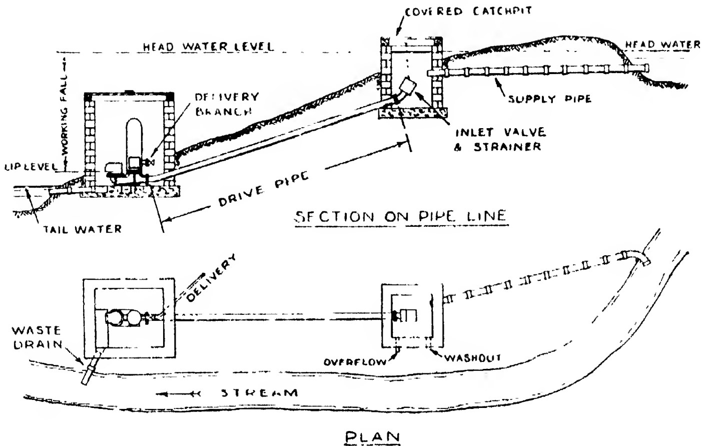


FIG. 66.—LAYOUT FOR HYDRAULIC RAM

The general arrangement of an air-lift plant is shown in Fig. 65. It usually consists of an air compressor of single-stage type for pressure below 60 lb. per square inch and two-stage for pressure above 60 lb.

The following figures give the sizes of borehole, rising main, and air pipe, and the probable output per minute. The area of the rising main should be about six times the area of the air pipe, and the submergence not less than one and a half times the lift.

TABLE IV—BOREHOLE, RISING MAIN, AND AIR PIPE SIZES

Diameter of Borehole In.	*Diameter of Rising Main In.*	*Diameter of Air Pipe In.*	*Gallons Per Minute*
4	1¼	¾	25
4½	2	1	50
5	2½	1	75
6	3	1¼	100
7	3½	1½	150
8	4	1½	200
9	5	2	300
10	6	2	450

These deliveries are average, and depend to a very large extent on the nature of the water-bearing strata.

The air pipe should be outside the rising main if there is room in the borehole, as otherwise the friction loss is much increased and the output reduced. By flaring the lower end of the rising main, so that the water is started from rest gradually, considerably less power is required, and a gain is to be had also if

the air is directed upwards as it enters the rising main as shown in the diagram. A long, gradual taper at the delivery end of the pipe will also increase efficiency. When the water has to be delivered some distance horizontally, it must be discharged into a tank high enough to let it flow to the place required by grav.ty, or, alternatively, it may be handled by a separate pump.

Tests carried out by Alley & MacLellan, Ltd., have given the following results:

TABLE V.—SUBMERGENCE AND AIR REQUIREMENTS FOR VARIOUS LIFTS

Lift in Feet	*Volume of Air to 1 ft. of Water*	*Submergence equals 60 per cent Total Rising Main* Ft.	*Submergence for Special Lift* Ft.	*Air Pressure Submergence 60 per cent.* Lb.	*Air Pressure for Special Lift* Lb.	*Cubic Feet Free Air per Gallon per Minute*	*H.P. per Gallon per Minute*
25	2·0	38	50	17	20	0 3	0·0184
50	3·0	75	100	33	44	0·4	0·0426
75	4·5	113	150	49	65	0·6	0·0828
100	6·0	150	200	65	87	0·8	0·1320
125	7·5	188	250	82	109	1·0	0·1910
150	9·0	225	300	98	120	1·2	0·2544
175	10 5	263	350	115	152	1·4	0·3150
200	12·0	300	400	130	173	1·4	0 3808

THE HYDRAULIC RAM

When conditions are suitable, the hydraulic ram provides a reliable and most inexpensive method of raising water to a considerable height for domestic and industrial purposes. It is essential that a constant head of water is available from a spring, reservoir, or stream, in order to obtain the necessary momentum to work the ram.

It will have been noticed that when water under pressure is flowing freely through a water tap and the tap is suddenly closed, water hammers or shock, of considerable magnitude, usually occurs. If the tap is then opened and the water allowed to gain velocity, the shock will be repeated when the tap is again closed. It is this repeated shock action which operates the hydraulic ram.

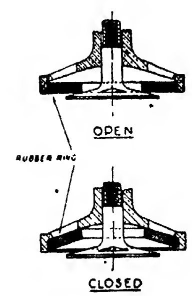


FIG. 67.—PULSE VALVE USED IN RAM

A typical layout is shown in Fig. 66. The action when the ram is in operation is as follows: water flows from the stream or head water to the catchpit, which acts as a reservoir and ensures a constant supply of water to the ram, then through the drive pipe to the inlet of the ram. A pulse, or waste valve in the ram shown in Fig. 67, remains open until the required water velocity is obtained, when it automatically closes, and the momentum, or energy, causes the water to lift a delivery grid valve and

overcome the pressure due to the head of water in the delivery main.

The quantity of water discharged by a hydraulic ram depends upon the working fall, the quantity of water flowing to the ram, and the total head against the ram, including friction in the pipes. Approximately one-seventh of the volume of water flowing into the ram can be raised to about five times the height of the working fall. A general rule is: multiply the water in gallons per minute flowing into the ram by the working fall in feet; multiply the product by 40, and divide by the static head against the ram in feet. This will give the delivery in gallons per hour.

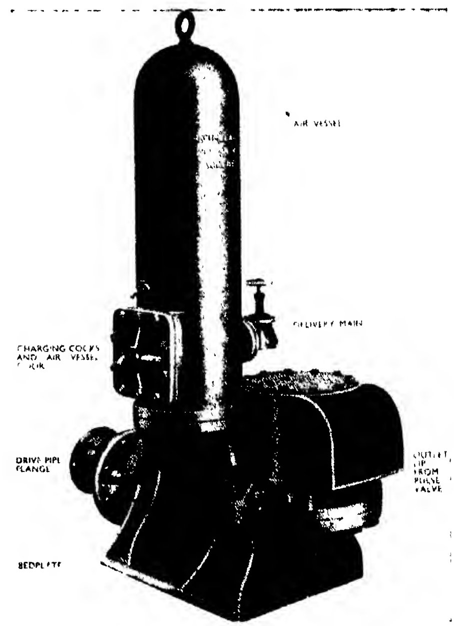


FIG. 68.—THE VULCAN HYDRAULIC RAM

Installation

The ram should be bolted firmly to a concrete bed, and care should be taken to keep the drive pipe straight and on an even gradient throughout. A given length to suit the size

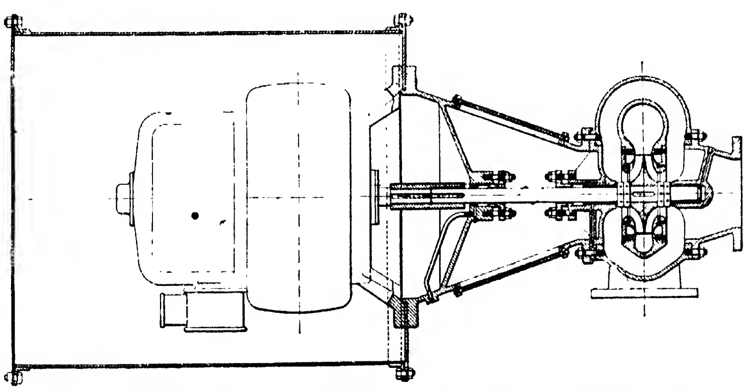


FIG. 69.—SECTION THROUGH THE MOTOR CASING AND PUMP OF A SUBMERSIBLE UNIT
(*Merryweather & Sons, Ltd*)

of ram must be strictly adhered to, and every joint must be perfectly sound and airtight.

The supply pipe should be large enough to carry the required quantity of water to the catchpit without loss of head, and the rising main must have an open end at the highest point, where an overflow must be fitted.

The air valve just below the air vessel must always be above water. The Vulcan ram shown in Fig. 68 is manufactured by Messrs. Green & Carter, Ltd., in eleven standard sizes. The smallest, requiring 1 gallon to 4 gallons of power water per minute, raises 100 gallons to 400 gallons per twenty-four hours, and the largest, with a 10-in. drive pipe and requiring 300 gallons to 400 gallons of power water per minute, discharges from 2,700 gallons to 4,000 gallons per hour.

SUBMERSIBLE SALVAGE PUMPS

The first apparatus of this class was adopted for pumping out wells and flooded mine shafts, where an appliance was necessary that could be supplied with power from the surface and lowered down to considerable depths, following the water at its level, due to the action of the pump, and that could not be put out of action by a sudden rise of the water level, from whatever cause.

The essential features of such a pumping appliance were an electric motor of standard type that could take power from either a continuous or alternating-current supply, as the case might be, fixed in a hermetically sealed steel cylinder, and a pump either of reciprocating or centrifugal pattern, fixed in or outside the cylinder itself, the driving shaft of which was arranged in such a manner as to prevent any possible ingress of water to the cylinder through the stuffing boxes in which it revolved.

The pump shown in Fig. 69 is a standard equipment that can be operated from lighting or power plant, and readily put to work in a confined or flooded compartment. The unit illustrated has a capacity of about 69,000 gallons per hour and has Merryweather standard centrifugal-type pumps. The electric motors are of 20 b.h.p., and are arranged so that they will work equally well in a horizontal or vertical position. They are fixed in galvanised-steel cylinders of stout construction to withstand rough usage, lined with non-conducting composition, and having a manhole at one end closed by a cover with rubber jointing ring and wing nuts. The motors are designed for a 220-volt continuous current. The pump and motor spindles are connected through a patent intercepting chamber, having a stuffing box at each end and bolted covers for gland packing and adjustment.

The pumps have cast-iron chambers with gunmetal impellers and bronze spindles, and have suction and delivery branches 8 in. diameter, with swing-bolt couplings to British Admiralty standards for the attachment of flexible piping.

Another type of submersible pump is illustrated in Fig. 70. This is manufactured by Messrs. Hayward-Tyler & Co., Ltd., in two standard sizes capable of dealing with from 99 tons to 280 tons of water per hour with heads up to 75 ft.

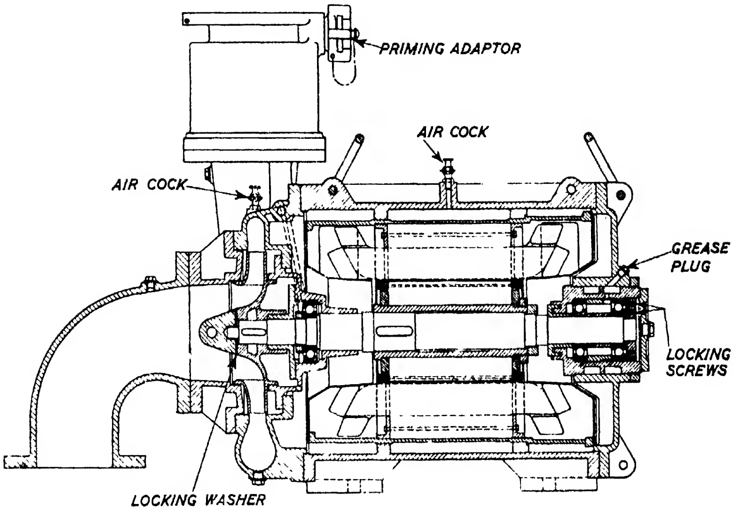


FIG. 70.—SECTION THROUGH A SUBMERSIBLE PUMP
(*Hayward-Tyler & Co., Ltd.*)

The impeller is of gunmetal, proportioned to develop the specified head with the maximum efficiency. It is keyed to the shaft and secured against a shoulder by a nut. It is not necessary to remove the impeller before withdrawing the pumpshaft and rotor.

The shaft is carried by a roller bearing at the pump end and two ball-thrust bearings at the motor end. The shaft is also the motor spindle, and is threaded for locknuts securing the rotor, the impeller, and the two bearing assemblies.

The laminated stator is wound with P.V.C. cable. This is impervious to water, a filtered supply of which flows constantly through the windings for cooling purposes. A gunmetal shell provides a sufficiently resilient mount for the stator to absorb any expansion due to abnormal operating conditions, and also facilitates the removal of the stator from the motor housing.

The rotor is built up from laminations pierced to receive the copper bars forming the squirrel cage. The assembly is mounted between end plates on a cast-iron sleeve, which is keyed to the shaft and secured against a shoulder by a locknut.

E. P.
E. M.

SURVEY OF STEAM BOILER PLANT

STEAM boilers of relatively large capacity may be divided broadly into two basic types, water-tube and fire-tube boilers; while there is also the electrode boiler, which, in recent years, has become popular with users of relatively small quantities of steam. According to the boiler design, the steam pressure generated may be anything from a few pounds per square inch up to 3,226 lb. per square inch, which is the critical pressure or maximum at which steam can be produced.

As the name implies, in water-tube boilers water is contained within a series of tubes, and is heated by hot gases surrounding the tubes. In this type of boiler the transmission of heat is rapid and overheating is prevented; while steam can be produced in a comparatively short time, because the water in the boiler is divided up into small streams, each of which is entirely surrounded by hot gases.

Water-tube boilers may be either of the vertical or horizontal pattern. In the fire-tube class of boiler, the hot gases from the furnace pass through the tubes which are surrounded by the water. The electrode boiler consists essentially of a series of electrodes dipping direct into the water to be boiled, so that the heat is generated within the water itself by the resistance offered by the water to the passage of the current. Electrode boilers have the advantage of not being subject to most of the losses encountered in the larger types.

Boiler Efficiency

Some of these losses are due to incomplete combustion of carbon, unconsumed combustible matter, radiation, moisture in the fuel and in the air fed to the boiler and to the formation of water from the combustion of hydrogen. In most modern boiler plants devices are in use to keep such losses to a minimum, while a check is kept on them by means of automatic instruments, and by technicians employed for that purpose. What is termed the "over-all efficiency" of a boiler is obtained by comparing the yield of B.Th.U.s in the steam produced with the B.Th.U.s in the fuel used. Such efficiencies often lie between 70 and 80 per cent., but in some cases they may be lower or even higher.

Evaporative Power

The evaporative power of any specific boiler depends on the grate area of the boiler and the heating surface. Grate area determines the quantity of fuel necessary, while the heating surface governs the amount of heat which the

boiler can absorb. So that with sufficient grate area, boiler power depends on the extent of the heating surface. The most economical design is obviously that having the largest area of heating surface in proportion to the bulk of the boiler when due allowance is made for steam space, and similar items.

TYPES OF FUEL

In Great Britain the most widely used type of fuel for boiler firing is coal, alternative types of fuel being coke, mineral oil, town's gas, and producer gas.

Coal

From the steam-raiser's point of view, coal consists of combustible and non-combustible matter, and the proportions of these present influence the calorific value of the coal, i.e. the number of B.Th.U.s obtainable from a given amount of the fuel. The combustible matter consists of carbon (coke) and volatile matter, while the non-combustible portion comprises ash and water. In purchasing such fuel in bulk, therefore, the boiler-plant manager is vitally interested in an approximate analysis of the coal, since this will indicate roughly its performance in the plant. The B.Th.U.s obtainable per pound of coal varies with the quality of the latter, and usually ranges between 11,000 and 14,000 B.Th.U.s. Coal in pulverised form is often used in grates designed for this type of fuel, while coke is also employed.

Mineral-oil Fuel

The use of mineral oil as fuel dispenses with the necessity for ash removal, while in many cases much labour is saved; but in Great Britain oil is generally much dearer than coal or coke, so that its application has so far been limited. Such oil is stored in tanks away from the boiler furnaces, and is fed to the latter by a pump. Two basic types of oil burners are employed. In one of these the necessary atomisation of the oil is brought about by the passage of the oil under pressure through a specially designed jet, while in the other one the oil mixes with a current of compressed air or steam which converts it into a fine spray. The heating value of the average fuel oil is much higher than that of a corresponding weight of coal, and is often around 20,000 B.Th.U.s per pound of oil. There is a certain similarity in feeding a furnace with oil and pulverised coal or coke, for all these are handled as liquid fuels, and in some cases a mixture of oil and pulverised fuel is used.

Advantages of Town's Gas

The ideal type of fuel is ordinary town's gas, but in the majority of cases, in spite of the special low rates obtainable, this fuel is too costly. Where, however, conditions are favourable to the use of gas, it has many advantages. Transport of the fuel costs nothing, there are no ashes to be removed, while gas is a versatile fuel which can respond quickly to any variations in steam demand. With

gas it is possible to use all the heat units available, whereas with solid fuels some of the heat is invariably wasted. Such advantages to a certain extent offset the initial high cost of the gas. In certain places the boilers may be operated with coke-oven gas obtained from an adjacent undertaking.

Producer Gas

A lower-grade type of gas which is also suitable as a boiler fuel is the so-called "producer" gas, while waste gas from forges, rolling mills, and oil engines is also utilised. In many instances these by-product types of fuel are available only at intervals, and to keep a constant supply, use is made of an oil supply which can be switched on or off according to the amount available.

AUXILIARY EQUIPMENT

In addition to the boiler and its furnace, other essential parts of the plant include feed-water pumps, injectors, economisers, and superheaters.

Feed-water Pumps and Injectors

Both feed-water pumps and injectors serve the purpose of keeping the boiler supplied with water, and in installations of appreciable size it is usually advisable to have both available for the supply of water, as a safeguard in case of failure of one of them. Apart from this consideration, an injector is often the most economical and appropriate apparatus for feeding a small boiler, while a pump is generally most suitable for larger installations. The average injector will lift water to a height of a few feet, but they usually function most efficiently when there is a head of water, while the latter should be cold. The type of feed pump most commonly used is a duplex steam pump, but centrifugal pumps and triplex pumps are employed to a limited extent, these being generally motor-driven.

Preheaters

Several methods are in use for preheating the water entering the boiler, and they may be divided into two basic types. One of these is the open-type heater, wherein the heating medium and the water contact each other. The disadvantage of this method is that—although heat transfer is rapid—impurities may be introduced into the water and so pass into the boiler. A closed-type heater, or an economiser, is frequently employed, and this precludes the introduction of any impurity into the feed water, for the latter passes through a series of pipes surrounded by the hot gases from the furnace. As these gases often have a temperature of at least 600° F., the heat is put to a useful purpose instead of going to waste up the chimney.

Superheaters

The hot furnace gases also serve in many cases another useful purpose, to raise the temperature of the steam which has been generated by the boiler,

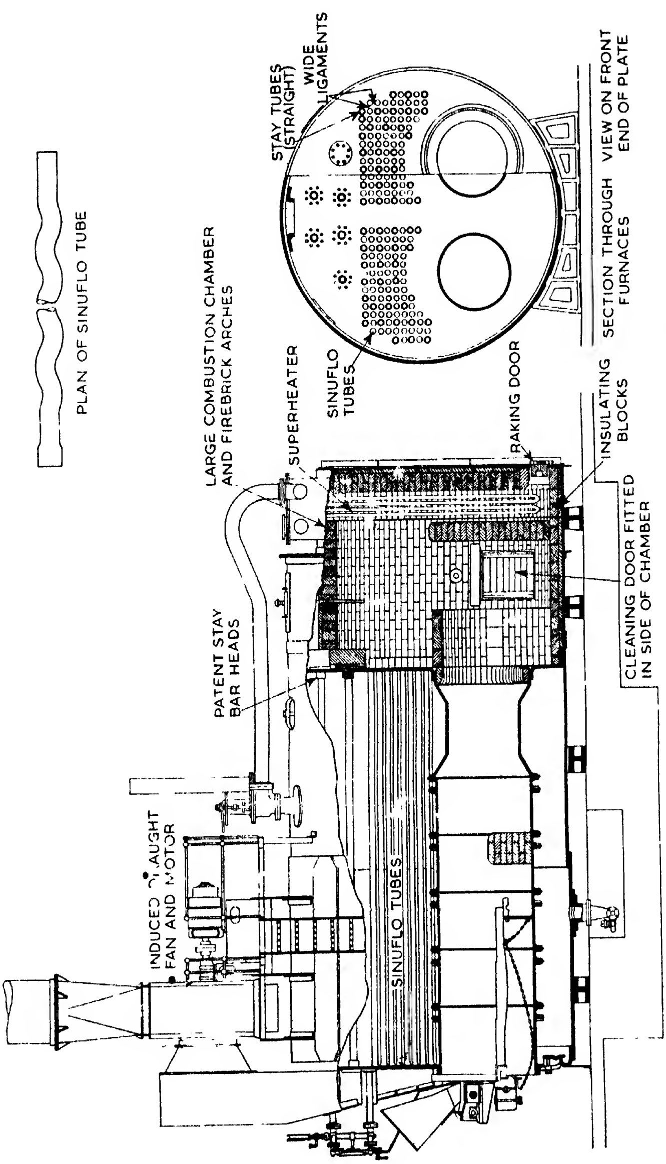


FIG. 1.—DIAGRAMMATIC ARRANGEMENT OF THE COCHRAN "SINUFLO" ECONOMIC BOILER

generally termed superheating. This has several advantages, for it obviates an abnormal amount of condensation in the steam lines from the boilers to the points of utilisation; it enables quicker heating and higher temperatures to be obtained in process heating, as in chemical works; both engines and turbines operate more efficiently when the steam is superheated, while superheated steam does not give up its heat so rapidly in passing through the pipe systems. Many types of superheaters are in use, and they are designed to suit the boiler to which they are attached. A typical example is the Yarrow superheater, which consists of a hollow forged drum, into which groups of U-tubes are expanded.

Air Heaters

To ensure adequate combustion of the fuel, it is necessary to force air into the furnace in amounts much greater than that theoretically required for complete combustion; and so that the air current shall not create an undue loss of heat within the furnace the air is heated, this heat being obtained from the flue gases. A common type of air heater comprises a pair of tube plates joined by straight tubes through which the hot gases pass, while the air flows around the outside of the tubes.

It will be obvious that if the air entering the heater is very cold, the heater may act as a condenser to condense certain constituents in the flue gases, which may result in dilute sulphuric acid attacking the metal surfaces. To prevent this, the air entering the heater should have a minimum temperature of 90° F., and in humid climates a higher temperature is necessary to prevent condensation of moisture on the metal. In operation the heater may be so adjusted that the temperature of the air entering, added to the temperature of gases leaving the heater, never falls below a specific figure. This is achieved by adjusting dampers for air and the use of by-passes for the recirculation of the air.

BOILER CONSTRUCTION

In the construction of boilers, both riveted and welded work is employed. In the case of welding, the seams are examined by X-rays in order to detect any possible flaws in the welded work.

Land Boilers

A typical example of a modern boiler is the Cochran "Sinuflo" Economic type, illustrated by Figs. 1 and 2. This is of the horizontal multi-tubular dry-back type, the standard design operating with induced draught generated by a fan. The boiler shell is made with two strakes or belts, and the mid-circumferential seam is treble-riveted. It is fitted with a single pass of tubes, the high transmission efficiency of which makes it unnecessary to fit a double pass.

This high efficiency is due to the wavy pattern of the tubes which causes the gases passing through them to impinge upon the metallic surface. In a straight tube a core of gases may pass through without coming into actual contact with the tube wall. The tubes are swelled at one end, are inserted through the front

tube plate in the same manner as a straight tube, and then expanded in the tube holes. The waves in the tube structure are in a horizontal plane, so that the bottom of the tube is level, thus avoiding dust pockets.

A relatively large brick-lined combustion chamber is fitted, and projecting into it from the mouths of the furnaces are brick arches, which compel the hot gases to travel round the combustion chamber before entering the tubes. Thus both space and time is given for complete combustion, which is materially helped by the large mass of hot brickwork. The reduction in the velocity of the gases, combined with the centrifugal action created, causes a large proportion of the grit entrained in the gases to be deposited in the bottom of the chamber, from which it is easy to remove.

In this type of boiler the stay bars are screwed into bosses which are riveted to the outside of the tube plates, so that no stay-bar threads pass through the tube plates, as this latter arrangement would be a potential source of leakage. Evaporative capacities of these boilers range from 6,000 to 30,000 lb. of steam per hour from and at 212° F. Their diameters range from 6·5 ft. to 13 ft., larger units being limited in size only by difficulties of transport. A battery of these boilers is seen in Fig. 3, installed in the Mossend, Lanarkshire, works of Imperial Chemical Industries, Ltd. The units are 9 ft. 6 in. in diameter, by 15 ft. 8 in. over the tube plates, operating at a pressure of 120 lb. per square inch, and giving a total output of steam of 54,500 lb. per hour from and at 212° F. Such boilers operate frequently without the aid of an economiser, and an efficiency of around 80 per cent. gross on the net calorific value of the fuel is normally obtained, irrespective as to whether the fuel is coal, coke, oil, or coke-oven gas.

FIG. 2.—A TYPICAL EXAMPLE OF A COCHRAN "SINUFLO" ECONOMIC BOILER

The boiler is 8 ft. in diameter, operating at a pressure of 125 lb. per square inch, and in this instance is fitted with a Hodgkinson's automatic stoker. (*Cochran & Co., Annan, Ltd.*)

Marine Boilers

In the construction of marine boilers more thought has to be given to the weight and size of the unit than is the case with land boilers, and on board ship the object is to provide a high-powered unit of relatively low weight and occupying a minimum of space. The Yarrow boiler is one of the best-known types of marine installation, and a typical example of this unit is outlined in Fig. 4. It may consist of either two or three water drums connected by banks of generating tubes to one large steam drum.

All parts subject to internal pressure are circular in section, and the generating tubes are straight or nearly so. The weight of these parts is carried either on feet attached to the lower drums or on a cradle formed by the casing; in both methods suitable sliding contacts are used to minimise stresses due to differential expansion. Since the cross-sectional arrangement of the drums and tubes does not vary along the length of the boiler, the possibility of differential stresses under the action of heat is further reduced. Due to these low stresses, the boiler is of light weight, and is suitable for operation at high pressures.

The drums used in the Yarrow boiler are frequently solid forged, with ends welded on or closed in, but in some instances they may be rolled from steel plate, with the longitudinal joint made by welding, the only bolted joints being the manhole covers. With the higher steam pressures and temperatures employed to-day, the absence of riveted and bolted joints is a desirable feature, especially from considerations of upkeep and the prevention of caustic embrittlement. The large size of the steam drum provides sufficient water surface area and steam space to ensure that steam of suitable quality is passed over to the superheater. This is an important feature, particularly when the water, and hence the steam, is liable to be contaminated, for it prevents impurities being carried over

FIG. 3.—A BATTERY OF COCHRAN "SINUFLO" BOILERS INSTALLED IN THE MOSSEND, LANARKSHIRE, WORKS OF IMPERIAL CHEMICAL INDUSTRIES, LTD.

The boilers are 9 ft. 6 in. diameter by 15 ft. 8 in. over the tube plates, operating at a pressure of 120 lb. per square inch, and producing 54,500 lb. of steam per hour from and at 212° F. (*Cochran & Co., Annan, Ltd.*)

and deposited in the superheater tubes. The drums are all located fore and aft in the ship to reduce the effect of surging in a seaway.

In this type of boiler there are normally only two sizes of generating tubes, the larger size being used to form the highly rated radiant heating surface, and the smaller for the convection banks. Tubes enter the drums at an angle, sufficient parallel rolling contact being available to ensure an efficient joint. This arrangement allows the tubes to be disposed to suit the requirements of heat transfer without the use of awkward bends. The tubes are expanded in the tube plate in the usual manner, belled, and then re-expanded. The diagram on page 242 shows the compact layout on board ship with this type of boiler. Note the short

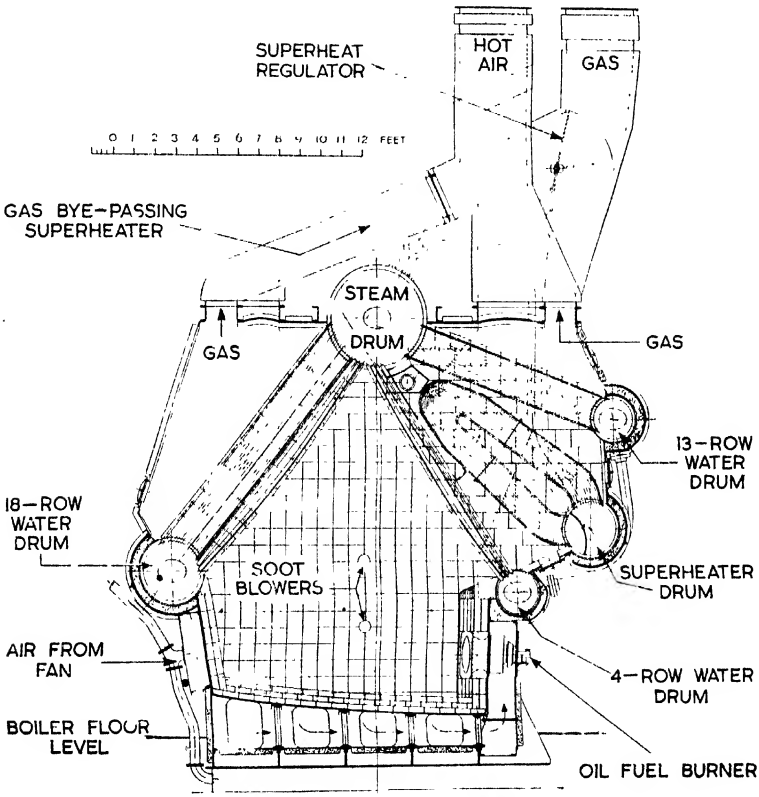


FIG. 4.—YARROW 5-DRUM MARINE BOILER, FITTED WITH A YARROW SUPERHEATER

The boiler is of the double-flow type, is side-fired, and is fitted with a Howden-Ljungström air preheater. Normal output is 68,000 lb. of steam per hour at a pressure of 475 lb. per square inch and a temperature of 750° F., from feed water at 320° F. (*Yarrow & Co., Ltd.*)

overall length of the stokehold and the wide central firing aisle. This unit is of the five-drum, double-flow, side-fired type, being fitted with Yarrow superheaters and tubular air heaters.

Mobile Boilers

In common with the marine boiler, the mobile type of steam boiler has to be designed to ensure a minimum of weight and size. An example of such a unit is the Fraser mobile vertical-drum water-tube boiler, seen in Fig. 5. This model is fitted with burners for petroleum-gas fuel, feed pump, and feed-water tank, this tank containing a heating coil connected to a pump exhaust steam pipe, a steam jet in the short folding chimney giving the required draught. The unit illustrated is used for fieldwork on oil pipelines at the wells of a large oil company, and may truly be said to represent a complete boiler plant on wheels.

The Fraser boiler is made in three main parts: the steam drum, the tube bundle, and the combustion chamber. Even when the unit is much larger than the mobile type, the plant is always transportable, for the parts can be readily assembled on site and in positions which might be inaccessible to shell types of boilers. This unit can be constructed with either a vertical or a horizontal drum, an example of the latter type being seen in Fig. 6. The horizontal steam and water drum has a larger water capacity, and may be considered more suitable for process work; while the vertical-drum type occupies much less floor space.

The steel casing enclosing the combustion chamber is a support for the pressure parts, and can be constructed to accommodate any type of grate and fuel. Thus, for coke or anthracite a larger grate area is required than for bituminous coal. The tube bundle is fixed directly above the combustion chamber, with the water tubes set at a steep angle, so allowing—with accelerating circulation as the water is heated—the ready passage of the water and steam bubbles to the steam and water drum. Heat by convection, added to that by radiation from the incandescent fuel, is thus absorbed by the water while passing through the tube bundle to the steam and water drum. This drum is of large capacity, has a suitable disengaging area for the liberation of dry steam, and is connected to the tube bundle by main circulating pipes of large bore.

In this unit steam can be raised from the cold state to working pressure in about 30 minutes, due largely to the fact that unrestricted unidirectional water circulation is secured between the steam and water drum and the tube bundle.

Wet-back Boilers

A unit which is a development of the so-called wet-back marine boiler is the Ruston horizontal "Thermax" boiler. The wet-back combustion chamber used is said to be an improvement on the brick-lined chamber of other types. In this unit the products of combustion are in contact with water-touched heating surfaces from the firegrate to the boiler outlet, so that the heat generated in the furnace is transmitted to the water. Efficiency has been further improved by the incorporation of an additional bank of tubes, which results in the outlet tem-

perature of the flue gases leaving the boiler at the low figure of between 400° F. and 450° F., depending on the working pressure.

A diagrammatic arrangement of the boiler is seen in Fig. 7, its chief features being the wet-back combustion chamber and a double set of return tubes, while the first and second sets of tubes are fitted to tube plates at the back of the boiler, thus allowing for independent expansion. The tubes carrying cooler gases are arranged near the boiler shell and low down where the water is coolest, an arrangement which increases circulation and assists in heat transfer.

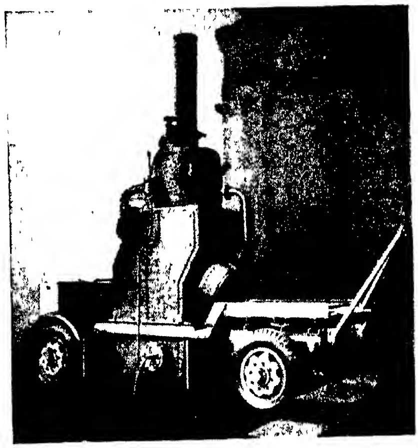


FIG. 5.—A MOBILE FRASER VERTICAL-DRUM WATER-TUBE STEAM BOILER

The boiler is fitted with burners for petroleum-gas fuel, feed pump, and feed-water tank; the tank contains a heating coil connected to the pump exhaust steam pipe. A steam jet in the short folding chimney gives the required draught.
(*Fraser & Fraser, Ltd.*)

When in operation, the products of combustion pass through the boiler flue to the wet-back combustion chamber, thence through the first pass of tubes to the smoke-box at the front, returning to the back of the boiler through the second pass of tubes to the outlet chamber. The set of fire tubes which takes the gases from the front smoke-box to the outlet chamber at the rear is divided into two nests, located on either side of the wet-back combustion chamber. This arrangement leaves the top of the boiler clear, thus giving ample steam space. It also avoids taking the products of combustion—when they are at their coolest point—through the top of the boiler where the water is at its hottest point. Moreover, the layout of tubes low down and near to the boiler shell

FIG. 6.—THE FRASER HORIZONTAL STEAM AND WATER DRUM

It has a large capacity, and is suitable for process work. (*Fraser & Fraser, Ltd.*)

assists the internal circulation of the water, and overcomes the tendency of the colder water to remain at the bottom where it is usually away from the heating surfaces.

Tubes comprising the first and second passes are arranged for independent expansion, thus eliminating the risk of tube leakage due to distortion of the tube plates caused by the unequal expansion of the two sets of tubes in one plate. The wet-back combustion chamber is cylindrical in form, and this allows space at the sides for the final set of tubes. Fig. 8 depicts a battery of five "Thermax" boilers, 10·5 ft. diameter by 13·5 ft. long, and one 9 ft. diameter by 12·5 ft. long, at Prestwick Airport. These boilers are built in standard sizes, with diameters ranging from 5 ft. to 12·5 ft., producing from 1,000 lb. to 14,000 lb. of steam per hour, from feed water at 60° F., and for working pressures up to 200 lb. per square inch.

Promoting Water Circulation

The heating of a volume of water of the magnitude often found within a boiler could easily lead to differential expansion of the shell plates—due to the

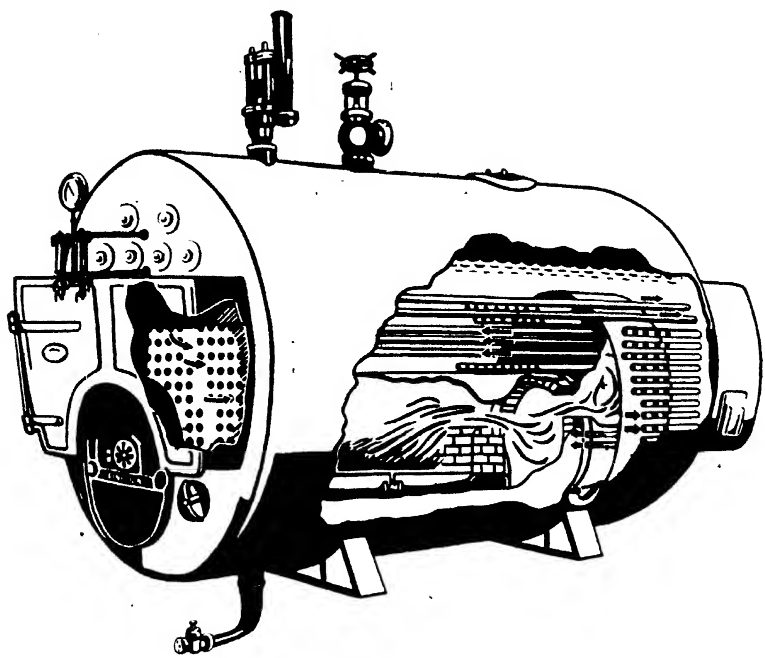


FIG. 7.—SHOWING THE PRINCIPLES OF OPERATION OF THE RUSTON "THERMAX" BOILER
Its chief features are a wet-back combustion chamber and a double set of return tubes.

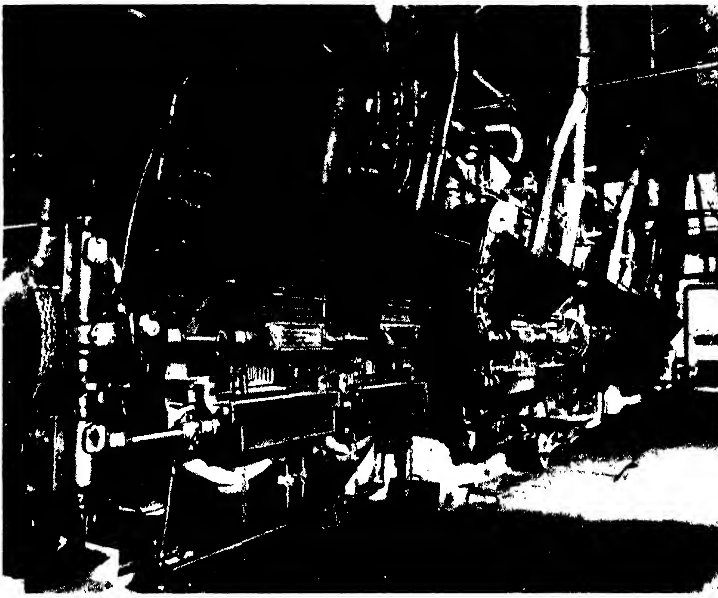


FIG. 8.—A BATTERY OF HORIZONTAL "THERMAX" BOILERS AT PRESTWICK AIRPORT

These are five boilers of 10·5 ft. diameter by 13·5 ft. long, and one of 9 ft. diameter by 12·5 ft. long. Boiler efficiencies up to 80 per cent. are obtained in this installation. (*Ruston & Hornsby, Ltd.*)

varying temperature in different parts of the water—if suitable provision was not made to avoid this. One way whereby this difficulty is overcome is aptly illustrated by the design of the Paxman "Ultranomic" boilers, Fig. 9 showing the double-pass type, and Fig. 10 the treble-pass type, of this boiler. In these boilers, owing to the hot gas passing through a bank of tubes close to the bottom of the shell, a free and vigorous natural circulation is promoted and sustained throughout the whole body of the water, thus eliminating dead spots. The single flue which is fitted is arranged on one side, not in the centre as is common practice, the tubes being positioned on the other side.

By adopting this form of construction, the furnace crown, where the greatest amount of heat is liberated, is raised much nearer the water level, thus allowing the steam bubbles to reach the surface easily. Because the tube nests are placed on one side, the lowest tubes are brought close to the bottom of the shell. This disposition of tubes and flue allows the fitting of a full-size manhole in the lower part of the end, thus facilitating inspection and cleaning. The gases, after passing through the flue, enter a combustion chamber located at the rear of the boiler. At the bottom of the chamber a rectangular pocket is formed which allows the

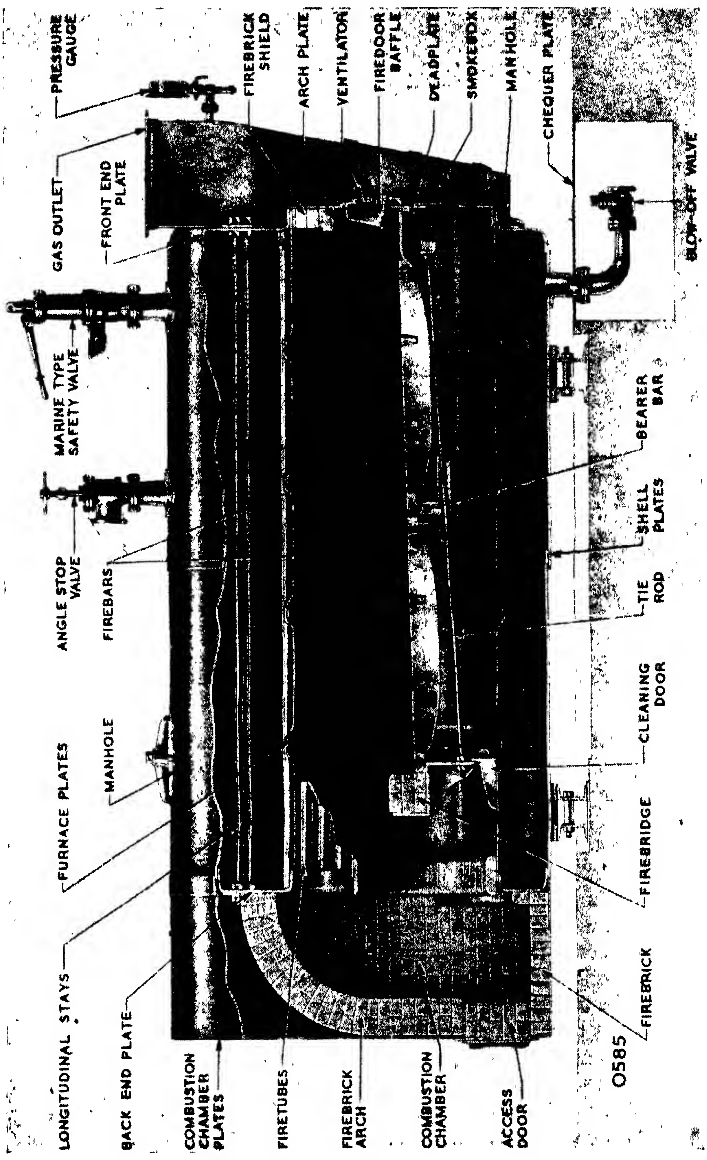


FIG. 9.—LONGITUDINAL SECTION OF THE PAXMAN DOUBLE-PASS "ULTRANOMIC" BOILER

accumulation of soot and ashes to be easily removed through the cleaning door.

On the inside surface of the firebrick-lined combustion chamber a high temperature is attained, thus igniting any gases which may have left the flue in an unburned condition. In this way better combustion is promoted, and at the same time the hot brickwork releases large quantities of radiant heat to the boiler back. The hot gases then pass to the front of the boiler through a bank of tubes. In these tubes, due to the high velocity and small cross-sectional area of the gas streams, a large amount of heat is given up to the water.

In the double-pass type the gases are collected in a smokebox and pass to the atmosphere via the chimney; while in the treble-pass boiler they enter a refractory-lined reversing chamber at the front of the boiler, from whence they are returned to the back of the boiler through an upper bank of tubes. Leaving the boiler at the rear, these gases are collected in the space over the combustion chamber and so pass to the chimney. In Fig. 11 is seen two treble-pass boilers of this type, 8·5 ft. in diameter. The amount of steam produced, in pounds per hour, varies with the size of boiler, and ranges between 1,800 and 13,302.

Waste-heat Boilers

In these days of high costs for fuels the utilisation of waste heat, whenever this is available, is an economic proposition. The application of waste-heat boilers to steelworks and other concerns has become an established practice, and permits of steam being raised in a very economical manner. Although modern open-hearth melting furnaces are provided with efficient regenerators, the gases leaving the latter have inevitably a temperature over 500° C. and this heat used to pass to waste up the chimney stack. About 44 per cent. of the heat content of the coal sent to the producers, which feed the melting furnaces with gas, may be dissipated to the atmosphere; but this figure is reduced to 22 per cent. if a waste-heat boiler is installed.

A 90-tons furnace, tapped, for example, fourteen times per week, will produce about 1,260 tons of steel, and over 1,750,000 lb. of steam can be raised from the waste gases. To raise this amount of steam in a coal-fired boiler would require nearly 100 tons of coal. It is obvious, therefore, that big savings in fuel are effected by using waste gases whenever possible for boiler heating.

The exhaust gases from internal-combustion engines are also sometimes put to use in a waste-heat boiler.

The "Sinuflo" Waste-heat Boiler

Some years ago much interest was aroused in the use of fire-tube boilers in this sphere, for research had shown the advantages of high-velocity gases for increasing the rate of heat transmission. When these gases are used in specially designed tubes, highly efficient results ensue, and the type of plant widely used for this purpose is the Cochran "Sinuflo" waste-heat boiler, an example of which is seen in Fig. 12; this boiler is 12 ft. in diameter by 15 ft. 8 in. over the

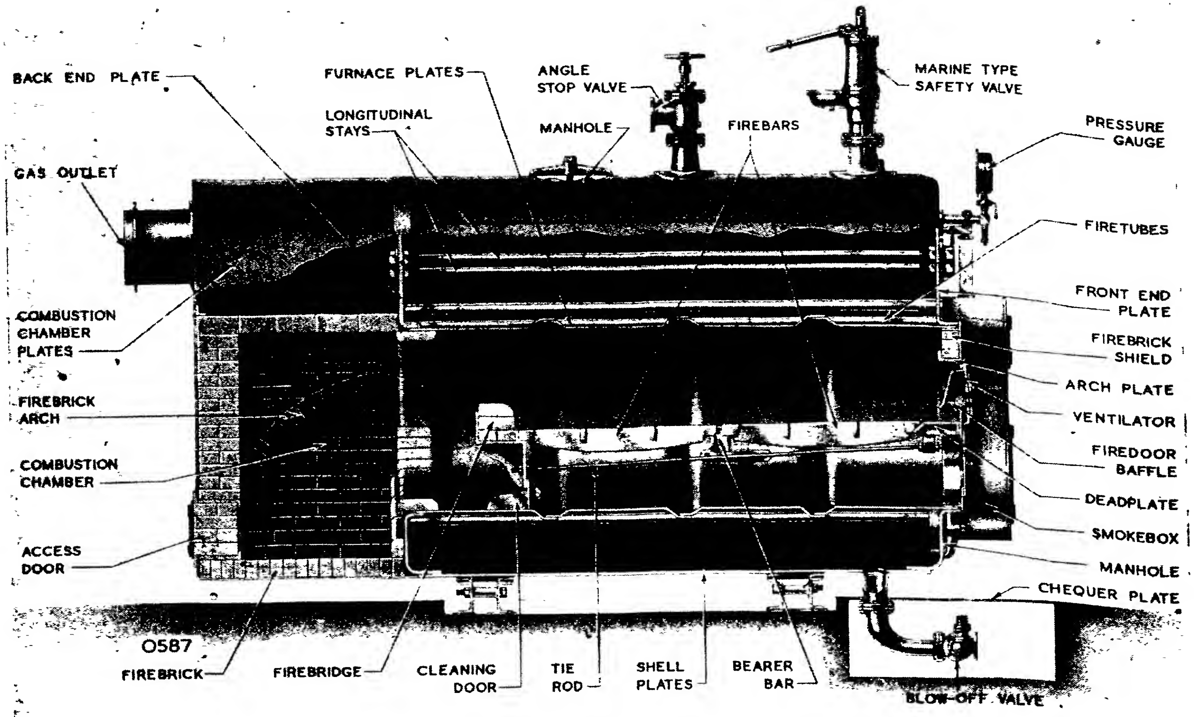


FIG. 10.—LONGITUDINAL SECTION OF THE PAXMAN TREBLE-PASS "ULTRANOMIC" BOILER
In these boilers, owing to the hot gases passing through a bank of tubes close to the bottom of the shell, good circulation of the water is promoted.

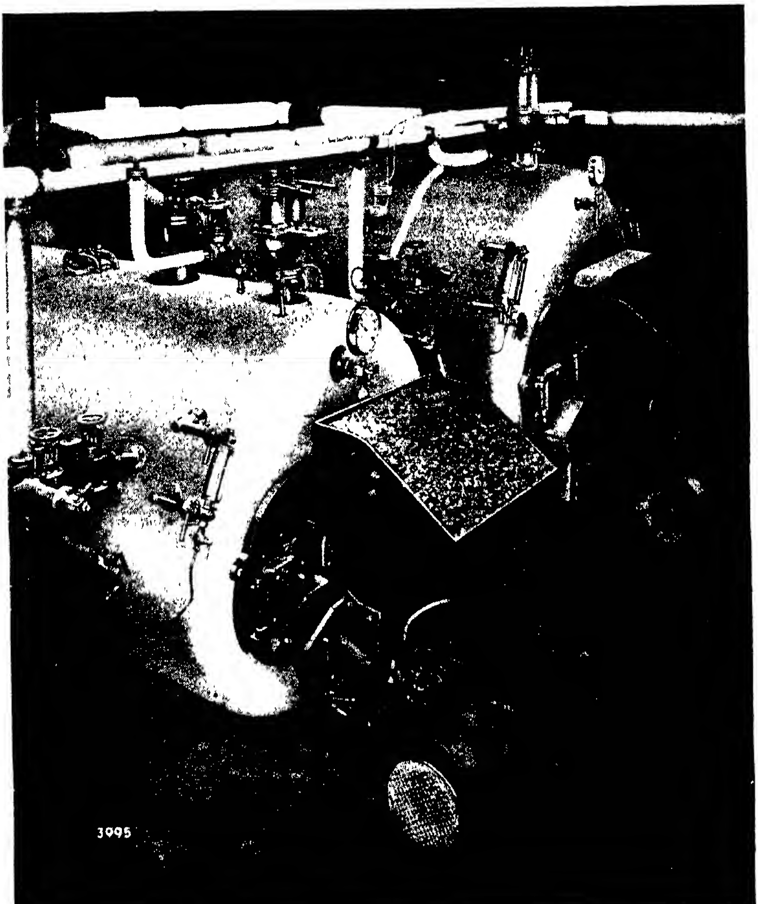


FIG. 11.—PAXMAN TREBLE-PASS "ULTRANOMIC" BOILERS, 8·5 FT. DIAMETER, IN THE WORKS OF WEST AUCKLAND CLOTHING CO., LTD. (*Davey Paxman & Co., Ltd.*)

tube plates. The gases approach the boiler through a steel inlet products chamber, lined with firebrick and fitted with explosion doors, a superheater being used when necessary. On leaving the boiler tubes, the waste gases pass through a mild-steel outlet products chamber to which is attached a cone piece forming the fan intake. An induced-draught fan is an essential feature of the installation, for it is extremely important to maintain the requisite draught at the

furnace throughout its operation. In addition, the fan must overcome the resistance through the flues and through the boiler, while a margin is also allowed to provide against adverse conditions.

The Use of Waste Gases from Furnaces

Steelworks furnaces are usually fired by producer gas, and the quality of the waste gases available can be ascertained by analysis of the coal originally used, the amount of coal fed to the producers, and the proportion of carbon dioxide (CO_2) in the waste gases. In some works blast-furnace or coke-oven gas is employed, and the value of the waste gases is calculated from their CO_2 content and the quantity and analysis of the gas supplied.

Evaporation in the boiler depends on (1) the volume or weight of gases available, and (2) the temperature of the gases as they enter the boiler. The curves, Fig. 13, show, for given percentages and temperatures of waste gases, how much steam may be raised per pound of coal fed to the producers. From such data one may calculate the amount of steam obtainable per ton of steel when the amount of coal used in treating the steel is known. One boiler is usually connected to one furnace, but the boiler can be designed to take the waste gases from two furnaces when required.

In a typical British installation the waste gases from seven open-hearth steel furnaces are led to seven "Sinuflo" waste-heat boilers, the furnaces being fired

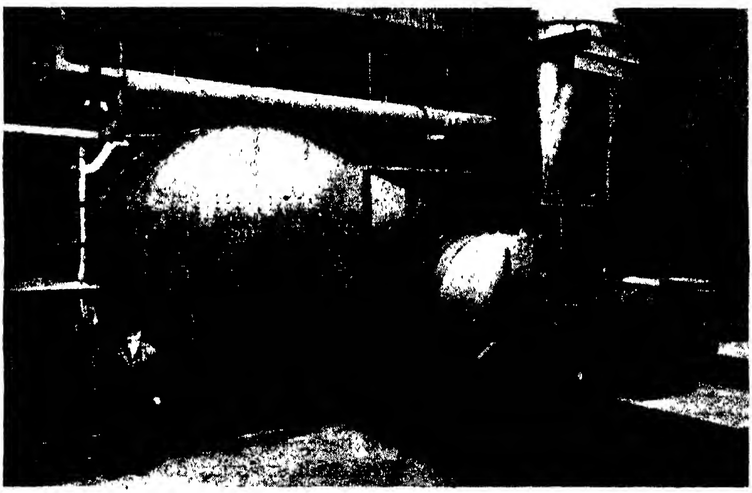


FIG. 12.—COCHRAN "SINUFLO" WASTE-HEAT BOILER, INSTALLED IN THE WORKS OF THE LANARKSHIRE STEEL CO., LTD.

This boiler takes gases from a 90-ton producer-gas-fired open-hearth furnace, and produces up to 12,000 lb. of steam per hour. (*Cochran & Co., Annan, Ltd.*)

with producer gas. Regenerators are installed in the gas and air ducts to the furnaces, the flow of gas and air is controlled, reversal of flow taking place about every 20 minutes. The waste gases, after leaving the regenerators, are led through underground flues to the waste-heat boilers. Short branches from the main flues lead to the inlet products chambers of the boilers, the flues to the stacks being closed by dampers. The waste gases at a temperature of 550–600° C. (1,022–1,112° F.) pass through the nests of "Sinuflo" tubes in the boilers to the outlet products chambers, thence through the fans and short independent funnels to the atmosphere. Superheaters, giving a final steam temperature of 520° F., are fitted in the inlet products chambers.

The boilers in this installation are 12 ft. 3 in. in diameter by 15 ft. 8 in. long over the tube plates, and are fitted with 538 "Sinuflo" and stay tubes $2\frac{1}{2}$ in. in diameter. The amount of steam produced by each boiler varies from 11,500 lb. to 14,500 lb. per hour, depending on the type of furnace supplying gases to the boiler; the amount of coal saved per week, by using the waste gases for boiler operation, is about 850 tons.

The Electrode Boiler

When relatively small amounts of steam are required, the use of the electrode boiler offers some advantages over the fuel-fired types, since fuel has not to be stored nor handled, and there are no ashes to be moved. The electrode boiler is designed on the principle that when an electric current is passed through water the temperature of the water rises. Electrical energy passing through the electrodes is converted 100 per cent. into heat within the water itself.

The boiler consists essentially of an earthed metal shell into which are fitted two or more electrodes. Current passes from one electrode to another, the water acting as an electrical resistance; thus heat is created. The quantity of heat generated is directly proportional to the square of the current and the resistance of the path. For a given voltage, the total heat produced is a function of the resistance of the water path and the immersed area of the electrode; so that the amount of steam raised in a specific boiler can be varied by simply exposing different areas of the electrodes to the water.

Operation of the Electrode Boiler

A diagram of a typical electrode boiler installation is seen in Fig. 14. In this, the feed pump supplies water to the base of the boiler shell, and when the water reaches a level required for the predetermined rating, the pump is stopped by a load-control relay (*CRI*) The effective immersed area of the electrodes is thus controlled to obtain a loading within about 5 per cent. of the required setting. By means of the rheostat (*RR*) the rating to give the required supply is adjustable within 30–115 per cent. of the specified equipment rating. Pressure is also predetermined by an adjustable setting, and when the desired pressure is attained, a pressure switch (*PS*) opens the magnetic valve, automatically adjusting the water-level, and proportionally the load, giving a constant pressure. This pres-

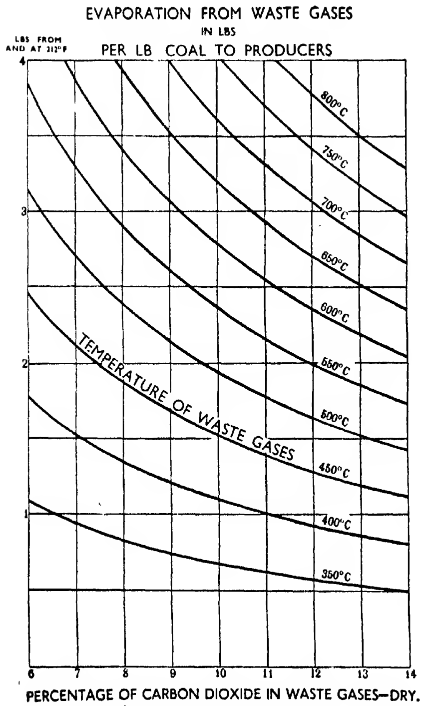


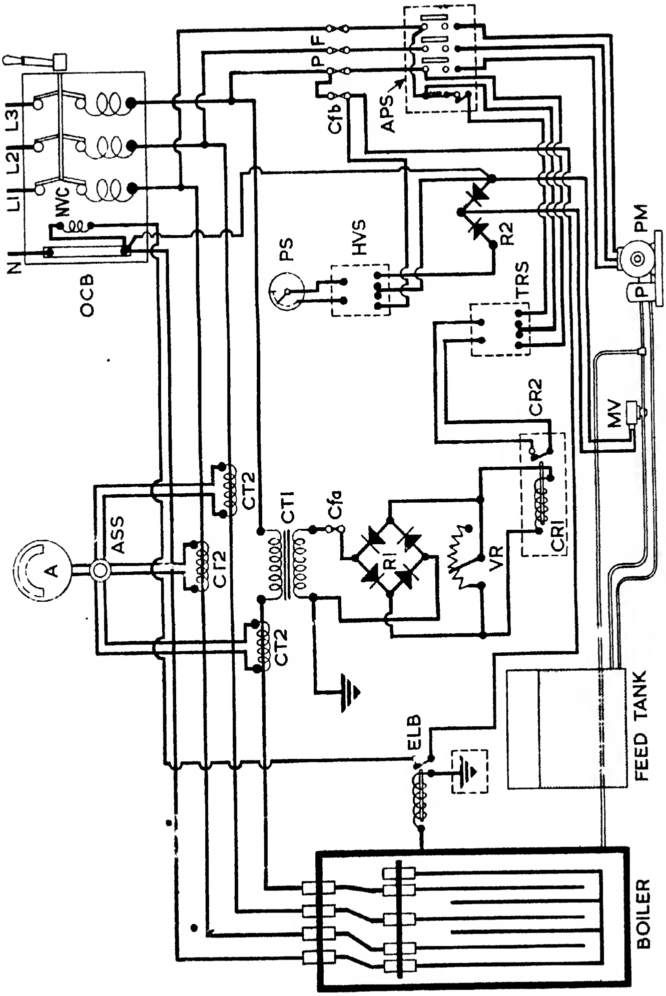
FIG. 13.—CURVES SHOWING, FOR GIVEN PERCENTAGES AND TEMPERATURES OF WASTE GASES, HOW MUCH STEAM MAY BE RAISED PER POUND OF COAL FED TO THE PRODUCERS
(*Cochran & Co., Annan, Ltd.*)

sure maintains instantaneous control within 1 lb. of the pressure setting. A hand-operated valve is fitted to the steam main to adjust the demand during operation.

It follows that should a high supply of steam be required, the water-level is controlled through the magnetic valve by the automatic load-control relay. Where supply requirements are reduced, the pressure switch takes control and automatically adjusts the water-level to suit the directly proportionate load. The entire unit, including feed tank, pump, control gear, and electrical equipment, is contained in a metal casing not much larger than a radio-gramophone. Capacities of the "Autolec" electrode boiler range from 30 kW., giving a steam output of 100 lb. per hour, to 1,000 kW., producing 3,350 lb. of steam per hour.

In Fig. 15 is seen a "B. & A." electrode boiler of 200 kW. capacity, which also operates fully automatically. The boiler is constructed of steel tube, with dished bottom and flat-top cover bolted to a flanged ring which is welded to the body. The electrodes and neutral shield are of cast-iron alloy, supported from porcelain bushings passing through the top cover, these being secured by packed steam glands of gunmetal. To prevent heat losses, the body and bottom of the boiler is lagged with magnesia asbestos blocks enclosed in a steel casing.

A large proportion of the electrode boilers in use operate at a steam pressure of 120 lb. per square inch, but no difficulty is experienced in making boilers for much higher pressures than this, and some of the "B. & A." models operate at 250 lb. per square inch. The electrode boiler is a very convenient means of raising steam in many different types of establishment where steam requirements do not warrant the installation of the bigger fuel-fired class of boiler. Such institutions as hospitals, restaurants, cooked-meat factories, dry cleaning works, and many other relatively small users of steam make use of the electrode boiler.



Key: OCB, Oil current breaker; *L1, L2, L3*, Lines 1, 2 and 3; *N*, Neutral; *NVC*, No-volt core; *A*, Ammeter; *ASS*, Ammeter selector switch; *CT1, CT2*, Current transformers; *PS*, Pressure switch; *PF*, Pump fuses; *PM*, Pump motor; *P*, Pump; *Cfa, Cfb*, Control fuses; *ELB*, Earth leakage breaker; *MV*, Magnetic valve; *VR*, Variable rheostat; *R1*, Full-wave rectifier; *R2*, Half-wave rectifier; *CR1*, Control relay coil; *CR2*, Contact on control relay; *TRS*, Time relay switch; *APS*, Automatic pump starter; *HVS*, Hot-wire vacuum switch.

FIG. 14.—SCHEMATIC ARRANGEMENT OF AN "AUTOLEC" ELECTRODE BOILER PLANT

The electrode boiler has the advantages of requiring only small space, no labour, and operates entirely automatically. (*G.W.B. Electric Furnaces, Ltd.*)

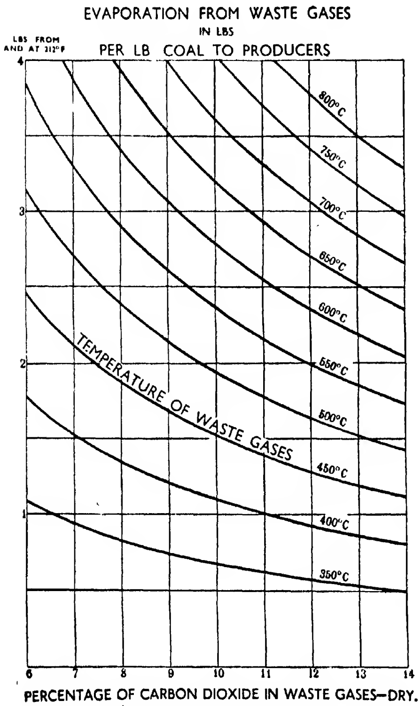


FIG. 13.—CURVES SHOWING, FOR GIVEN PERCENTAGES AND TEMPERATURES OF WASTE GASES, HOW MUCH STEAM MAY BE RAISED PER POUND OF COAL FED TO THE PRODUCERS
(*Cochran & Co., Annan, Ltd.*)

sure maintains instantaneous control within 1 lb. of the pressure setting. A hand-operated valve is fitted to the steam main to adjust the demand during operation.

It follows that should a high supply of steam be required, the water-level is controlled through the magnetic valve by the automatic load-control relay. Where supply requirements are reduced, the pressure switch takes control and automatically adjusts the water-level to suit the directly proportionate load. The entire unit, including feed tank, pump, control gear, and electrical equipment, is contained in a metal casing not much larger than a radio-gramophone. Capacities of the "Autolec" electrode boiler range from 30 kW., giving a steam output of 100 lb. per hour, to 1,000 kW., producing 3,350 lb. of steam per hour.

In Fig. 15 is seen a "B. & A." electrode boiler of 200 kW. capacity, which also operates fully automatically. The boiler is constructed of steel tube, with dished bottom and flat-top cover bolted to a flanged ring which is welded to the body. The electrodes and neutral shield are of cast-iron alloy, supported from porcelain bushings passing through the top cover, these being secured by packed steam glands of gunmetal. To prevent heat losses, the body and bottom of the boiler is lagged with magnesia asbestos blocks enclosed in a steel casing.

A large proportion of the electrode boilers in use operate at a steam pressure of 120 lb. per square inch, but no difficulty is experienced in making boilers for much higher pressures than this, and some of the "B. & A." models operate at 250 lb. per square inch. The electrode boiler is a very convenient means of raising steam in many different types of establishment where steam requirements do not warrant the installation of the bigger fuel-fired class of boiler. Such institutions as hospitals, restaurants, cooked-meat factories, dry cleaning works, and many other relatively small users of steam make use of the electrode boiler.

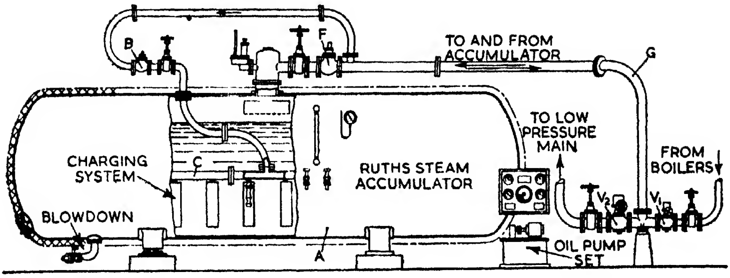


FIG. 16.—DIAGRAM SHOWING THE PRINCIPLE OF OPERATION OF THE RUTHS ACCUMULATOR
An accumulator increases the productive efficiency of a works, and improves the operation of the boiler plant by the elimination of peak-load firing. (*Ruths Accumulators* (*Cochran*), *Ltd.*)

If the boiler pressure tends to rise, due to a falling off in the general demand for steam, the valve (*V*1) opens and all surplus steam then being generated is passed to the accumulator by way of the pipe (*G*), where it is condensed and its thermal energy stored. If a peak load develops on the high-pressure main, the valve (*V*1) closes, and during the continuance of this condition, the steam demand of the low-pressure consumers is met by discharge of the accumulator. If a peak load develops on the low-pressure main, this is met by the wider opening of the valve (*V*2) and, when necessary, by the withdrawal of steam from storage to supplement the amount of steam then being obtained direct from the boilers through the valve (*V*1). The valves (*V*1) and (*V*2), therefore, automatically bring the accumulator into operation as a consumer of steam or as a producer of steam, as governed by the general variation in demand; while the boilers are fired at a constant rate equal to the average steam requirements in that particular establishment.

A typical Ruths accumulator installation is seen in Fig. 17, installed in a London establishment. In this instance the accumulator shell is 8·5 ft. diameter by 31·75 ft. long, giving a storage capacity of 6,200 lb. of steam over a pressure range of 110–25 lb. per square inch.

Feed-water Accumulators

There are many types of works wherein the main steam users operate at boiler pressure and conditions are unfavourable to the use of the steam-storage system. In these circumstances, the Ruths feed-water accumulator may be employed with advantage. The basic principle of this is that surplus steam is utilised for the building up of a reserve of preheated feed water, at or near saturation temperature; the peak-load carrying capacity of the system is therefore dependent on the permissible increase in feed temperature. In the majority of cases where the feed-water accumulator is applicable, peak loads of 25–30

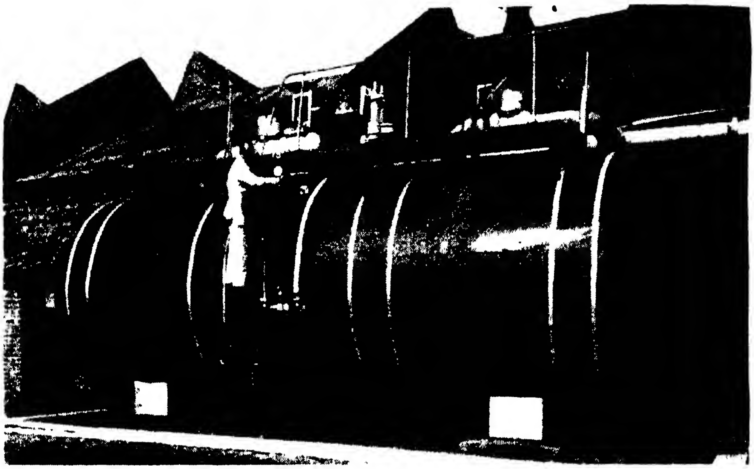


FIG. 17.—A TYPICAL EXAMPLE OF A RUTHS ACCUMULATOR INSTALLATION, AT THE ESTABLISHMENT OF A. BELL & SONS, LTD., LONDON

The accumulator is 8·5 ft. diameter by 31·75 ft. long, giving a storage capacity of 6,200 lb. of steam over a pressure range of 100–25 lb. per square inch. (*Ruths Accumulators (Cochran), Ltd.*)

per cent. in excess of the average demand may be met without a fall in steam pressure.

Another type of thermal storage involves the storage of hot process water, and this is very often applicable to the so-called process industries, such as laundries. The use of one system or another represents the only possible method of suppressing peak-load steam demands on the boiler plant and ensuring an adequate supply of steam to the consumers. Heat accumulation is accordingly an important principle in relation to economic production.

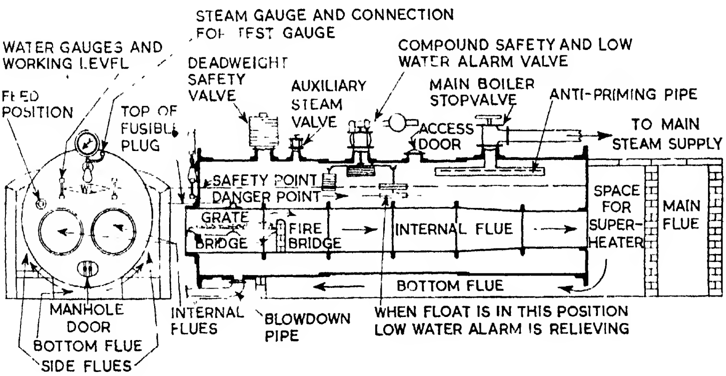
A. E. W.

BOILER-HOUSE WORK

BOILERS in common use in this country are usually of the shell or water-tube type, and are adapted in different designs to comply with the use intended. Nowadays the trend in power stations is to use water-tube boilers suitable for the higher pressures, and obviously higher superheated steam, and the pressures within recent years incline towards those nearing 1,000 lb. per square inch. Usually in this class of boiler-house work it is common practice to operate at 600 lb. per square inch working pressure. An increase beyond 600 lb. per square inch introduces problems in feed-water treatment and involves also the resistance of the metal parts to heat.

Industrial practice is concerned with both shell and water-tube types, and, of the former, may include the Lancashire, Cornish, Economic, Marine, and others of the shell class either separately or installed with the water-tube type in a large battery of boilers. The purpose served by the boilers, or steam generators, may be for generating electricity and heating, or process work, or a combined system of power, process, and heating in conjunction with steam accumulators, or by-pass turbines.

The turbine is favoured where steam is used for process work such as textile dyeing, as the lubricating oil does not come in contact with the steam,



FIGS. 1 AND 2.—SECTIONAL ARRANGEMENT OF A LANCASHIRE BOILER

Internally fired boilers should be fitted with a fusible plug or alarm device

whereas a reciprocating engine requires internal lubrication which, if a back-pressure machine, passes with the steam to the process machines.

Pressures

Pressures at which industrial plants operate when steam is generated may vary from 50 lb. per square inch to 600 lb. per square inch, according to the production plant, also whether with or without superheat. In some cases, dry-saturated steam will suffice, as the steam in this state imparts the heat to the process material or liquid during a shorter period than superheated steam.

A typical plant used for generating steam, electricity, and also supplying the exhaust steam to process work, may operate at 210 lb. per square inch and through reducing valves reduce the factory steam pressure to 40 lb. per square inch; consequently, machines such as drying cylinders may require 10 lb. per square inch pressure. Thus, by generating at a higher pressure, provision is made for electrical generation and process operation by stepping down the steam pressure. Obviously this entails a fairly elaborate system of protective devices, not only for safety but also to comply with the statutory regulations as laid down by the Factories Act of 1937. The saving in fuel, due to the use of higher pressures, can be set against the cost of any additional plant such as

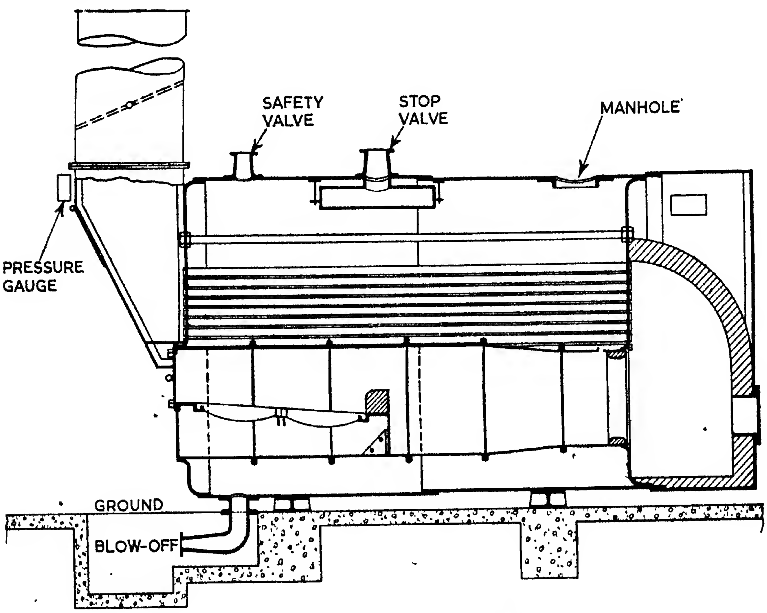


FIG. 3.—DRY-BACK ECONOMIC BOILER
(*Stockton Chemical Engineers and Riley Boilers, Ltd.*)

regulating, reducing, or safety valves to maintain the lower pressures. Furthermore, the expansive properties of the steam can be used to reduce steam and fuel consumption in prime movers such as turbines or engine-driven machines.

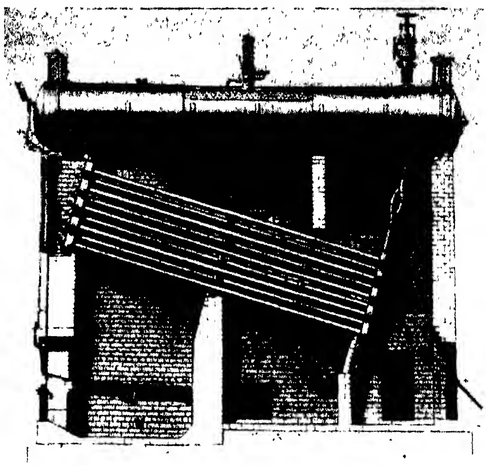


FIG. 4.—HAND-FIRED W.I.F. BOILER WITHOUT SUPERHEATER (*Babcock & Wilcox, Ltd.*)

Future developments and extensions, when under consideration, invariably become involved in the boiler capacity and working pressure. A heavy steam consuming factory is controlled by the output of the steam plant which must fulfil conditions to provide the maximum production, otherwise the steam services become interrupted, with undue production losses, when competing for deliveries.

To centralise the steam plant and operate at the higher pressure not only induces economy in operation of the plant, but in transmission of the steam through the steam mains to the process machines. The advantage gained is that as the latent heat in the steam decreases when the pressure is increased, the amount of heat required is proportionately less, then when the pressure is reduced the latent heat at the lower pressure can be recoverable; thus a two-fold gain is obtained.

To raise the pressure of one pound weight of steam from zero gauge pressure to 150 lb. per square inch requires approximately 49 B.Th.U.s and to raise it from zero to 350 lb. per square inch requires 61 B.Th.U.s; therefore only 12 additional B.Th.U.s are required for the increased pressure. The latent heat of steam at 300 lb. per square inch gauge pressure is 810 B.Th.U.s; at 15 lb. per square inch it is 946 B.Th.U.s, a difference of 136 B.Th.U.s.

Through the range of pressures from atmospheric pressure, the latent heat decreases as the pressure is increased. Less internal work in changing from the liquid state to the vapour state is carried out during the change. When 3,206·7 lb. per square inch is reached, no latent heat is contained in the steam, the total heat becoming 902·7 B.Th.U.s, which is also the sensible heat, and the temperature 705·4° F.

Raising Steam

When raising steam in a boiler it is some considerable time before the vapour stage is reached. The first indication that the boiler is absorbing heat from the furnace fires is particularly noticeable in the water-gauge glass as the level of the water rises, thus proving the density is decreasing and the water contained within the boiler is expanding. During this period the air valve, situated at the highest point of any boiler, remains open until the pressure is formed, thus expelling air and vapour. The valve is then closed after steam pressure is emitted.

As water is a bad conductor of heat, the process of steam raising cannot be hurriedly applied. The firing rate is slow. As the fires are charged, the dampers may be regulated accordingly. Should a heavy fire be introduced, the result is that the upper portion of the boiler becomes hotter than the lower; consequently severe strains and stresses are placed upon the metal shell and joints due to unequal expansion; this is inevitable when under steam. Should this practice be frequent, leaks occur in the joints and rivets are liable to be fractured. This applies more especially in the shell-type boiler and the vertical cylindrical class.

When raising steam, as for example at 40° F. temperature of the water in the boiler, it is good practice to bank fires for at least twenty-four hours before raising pressure if the boiler has not been in service for some considerable time. Then slowly work the fires until pressure begins to form. Consideration in this case must be of the refractories and brickwork. If major overhauls have

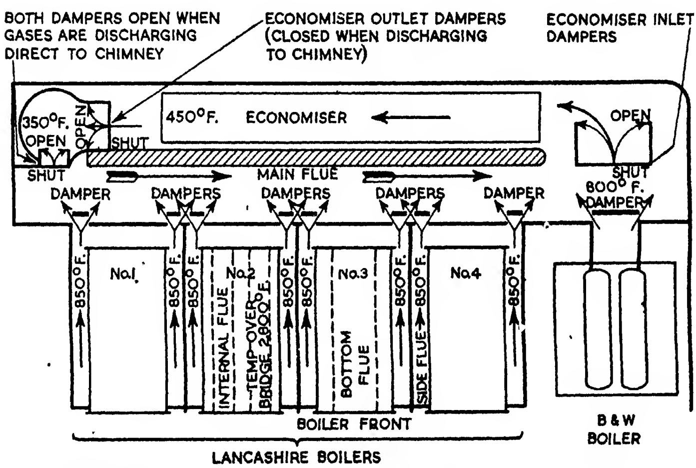


FIG. 5.—DAMPER SYSTEM ON LANCASHIRE AND BABCOCK & WILCOX BOILERS WITH ECONOMISER
Indicating temperatures throughout gas circuit.

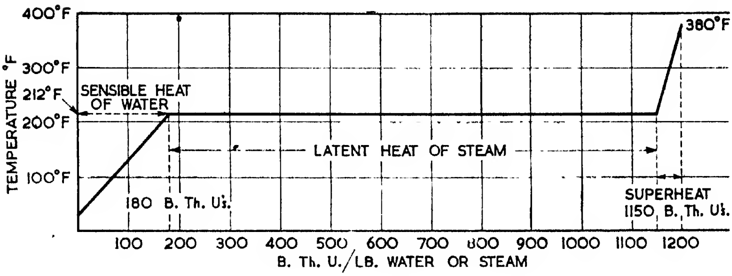


FIG. 6.—HEAT QUANTITIES EVOLVED IN THE FORMATION OF STEAM AT ATMOSPHERIC PRESSURE
(*Reproduced from "The Efficient Use of Fuel," by permission of the Controller of H.M. Stationery Office*)

been executed, three days are usually allowed to "dry out." Obviously, when overhauled, the water-gauge passages should have been cleared of any scale or obstruction. Thus, when under pressure, a true indication of the water in the boiler can be visualised when testing the water gauge to ascertain the exact water level in the water-gauge glass.

The specific gravity of water is unity when at its greatest density, approximately 40° F.; by applying heat by means of the furnace fires the temperature of the boiler water or liquid increases until boiling-point is reached. This heat is called "sensible heat," and the amount of sensible heat varies with the boiling-point of the steam, increasing towards the higher pressures. The density therefore decreases inside the boiler where heat is applied, and as the liquid increases in temperature the colder water follows, and thus circulation is created.

Precautions Necessary before Raising Steam

When filling a boiler with water it is not advisable to allow the water content to be at too high a level, in case, when steam is formed, it passes out of sight in the water gauge, thus creating a doubt as to whether the boiler is full or only partly filled.

It is essential that precautions are taken to ensure that dampers are free, also all valves operate, that the superheater is in communication with the boiler, and the blow-down valve is free.

Should the blow-down valve become obstructed or inoperable it may be necessary to empty the boiler to remedy the defect.

Sensible and Latent Heat

When vapour has formed and pressure has expelled all air from the boiler, the air valve can be closed, and the pressure allowed to increase slowly. Thus, if the maximum working pressure is 210 lb. per square inch, the amount of sensible heat required to raise the liquid to boiling-point, from Callendar's Steam Tables, will be 366·3 B.Th.U.s. The temperature at this pressure is 392° F., the latent heat is 839·2 B.Th.U.s, and the sensible heat plus the latent heat is

1,205 B.Th.U.s. Therefore, to reach boiling-point of the liquid internal work by heat supplied from the furnace fires is necessary to reach this point. If the pressure is forming, the rate of increase accelerates as indicated by the steam gauge. The nearer working pressure is approached, the greater the acceleration. This is due, not only to the heat absorbed from the fires, but to increased absorption of heat when passing through the range of pressures finally to working pressure, because the amount of latent heat required is less as a pressure increase takes place.

Some check or control of the rising steam pressure becomes imperative to prevent undue acceleration. If mechanically stoked, the firing rate is reduced and dampers closed in; should fan draught be used, the fans are stopped and natural draught applied. The following table illustrates the rise in liquid or sensible head and the decrease in latent heat from atmospheric pressure:

Gauge Pressure lb. per square inch	*Liquid or Sensible Heat B.Th.U.s*	*Latent Heat B.Th.U.s*	*Total Heat B.Th.U.s*	*Temperature of Steam ° F.*
Atmospheric 14·7	180·0	970·7	1150·7	212
50·0	266·0	915·0	1181·0	297
100·0	309·0	884·0	1193·0	338
150·0	338·0	862·0	1200·0	366
200·0	362·0	842·0	1204·0	388
210·0	366·0	839·0	1205·0	392
3206·2	902·7	0·0	902·7	705

The above figures apply to *saturated steam.*

Saturated and Dry-saturated Steam

When steam has absorbed the heat and is raised to the boiling-point until every particle of moisture has been evaporated, it is considered as saturated steam, and any loss of heat immediately commences condensation without a corresponding decrease in pressure. When steam contains no moisture, it is said to be dry-saturated, and when further heat is applied can rise in temperature after all particles of moisture have been evaporated. Latent heat enters into this consideration insomuch that it can be defined as the heat necessary to convert 1 lb. of water at boiling-point into dry-saturated steam to condense it all to water without any temperature reduction.

Superheated Steam

Until every particle of moisture in the steam is evaporated, it is regarded as saturated and passes to the dry-saturated state when the temperature rise takes place. This is practically the condition of the steam prior to entering the superheater. Various devices have been fitted to boilers for the purpose of entraining the moisture which may be in the steam before leaving the boiler, such as anti-priming pipes and baffles. These devices assist in reducing condensation in the main steampipe lines.

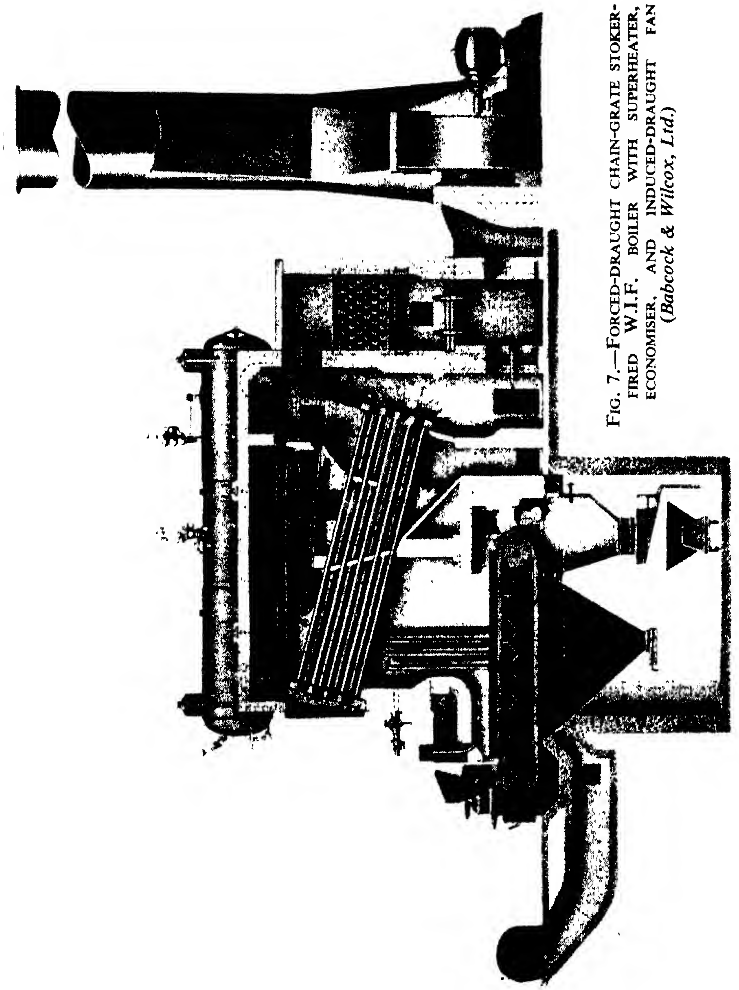


FIG. 7.—FORCED-DRAUGHT CHAIN-GRATE STOKER-FIRED W.I.F. BOILER WITH SUPERHEATER, ECONOMISER, AND INDUCED-DRAUGHT FAN (*Babcock & Wilcox, Ltd.*)

1,205 B.Th.U.s. Therefore, to reach boiling-point of the liquid internal work by heat supplied from the furnace fires is necessary to reach this point. If the pressure is forming, the rate of increase accelerates as indicated by the steam gauge. The nearer working pressure is approached, the greater the acceleration. This is due, not only to the heat absorbed from the fires, but to increased absorption of heat when passing through the range of pressures finally to working pressure, because the amount of latent heat required is less as a pressure increase takes place.

Some check or control of the rising steam pressure becomes imperative to prevent undue acceleration. If mechanically stoked, the firing rate is reduced and dampers closed in; should fan draught be used, the fans are stopped and natural draught applied. The following table illustrates the rise in liquid or sensible head and the decrease in latent heat from atmospheric pressure:

Gauge Pressure lb. per square inch	*Liquid or Sensible Heat B.Th.U.s*	*Latent Heat B.Th.U.s*	*Total Heat B.Th.U.s*	*Temperature of Steam °F.*
Atmospheric 14·7	180·0	970·7	1150·7	212
50·0	266·0	915·0	1181·0	297
100·0	309·0	884·0	1193·0	338
150·0	338·0	862·0	1200·0	366
200·0	362·0	842·0	1204·0	388
210·0	366·0	839·0	1205·0	392
3206·2	902·7	0·0	902·7	705

The above figures apply to *saturated steam.*

Saturated and Dry-saturated Steam

When steam has absorbed the heat and is raised to the boiling-point until every particle of moisture has been evaporated, it is considered as saturated steam, and any loss of heat immediately commences condensation without a corresponding decrease in pressure. When steam contains no moisture, it is said to be dry-saturated, and when further heat is applied can rise in temperature after all particles of moisture have been evaporated. Latent heat enters into this consideration insomuch that it can be defined as the heat necessary to convert 1 lb. of water at boiling-point into dry-saturated steam to condense it all to water without any temperature reduction.

Superheated Steam

Until every particle of moisture in the steam is evaporated, it is regarded as saturated and passes to the dry-saturated state when the temperature rise takes place. This is practically the condition of the steam prior to entering the superheater. Various devices have been fitted to boilers for the purpose of entraining the moisture which may be in the steam before leaving the boiler, such as anti-priming pipes and baffles. These devices assist in reducing condensation in the main steampipe lines.

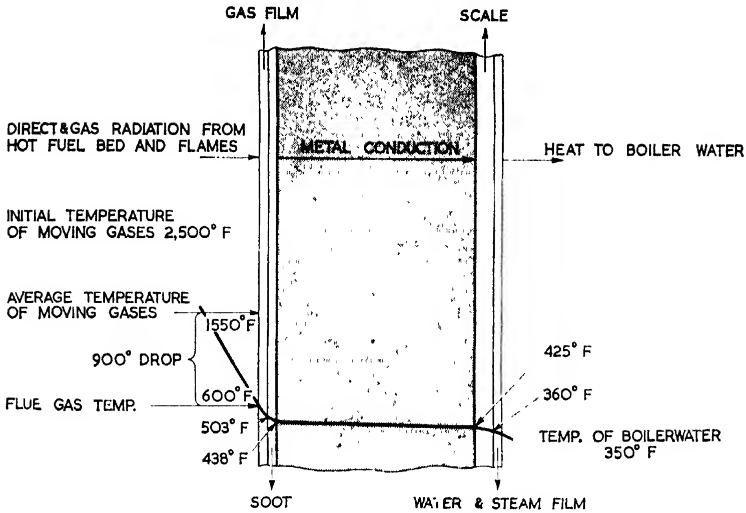


FIG. 8.—HEAT TRANSMISSION FROM HOT FLUE GASES TO BOILER WATER IN A BOILER TUBE WHEN EVAPORATING TEN TIMES NORMAL RATE

(Reproduced from "The Efficient Use of Fuel," by permission of the Controller of H.M. Stationery Office)

As the superheater is installed between the top row of circulating tubes and the bottom row of the economiser tubes, it receives the heat direct from the gases when raising steam. Each outlet is connected to a main steam header with a control valve, and a control valve at the end of each superheater outlet. Thus, when raising steam the latter valves remain open and also the drains. By this means there is a continual flow of vapour and steam and also heat transference which prevents overheating of the superheater tubes. A trap fitted at the end of a drop pipe in the main header discharges condensate taking place in the pipeline.

Steam can be raised quickly in this type of boiler, due to rapid promotion of the circulation. The lower tubes rapidly absorb the heat, whereby the buoyancy is produced by the heated water passing up the header to the drum and the opposite header water immediately receives a downward flow action. It is the refractories which must receive consideration, as also the boiler. In this respect the operation of steam raising cannot be contemplated hurriedly.

Therefore, after warming through with braziers, the fires can be made by burning coal or wood spread evenly along the front of each grate with the induced draught fan slowly revolving. For this purpose the guillotine door and the front air doors are opened, each coal hopper closed until the fire is properly ignited, and the arch heated, when the guillotine doors can be set to the correct thickness of the fire, usually 3–4 in., the induction-fan speed increasing as the

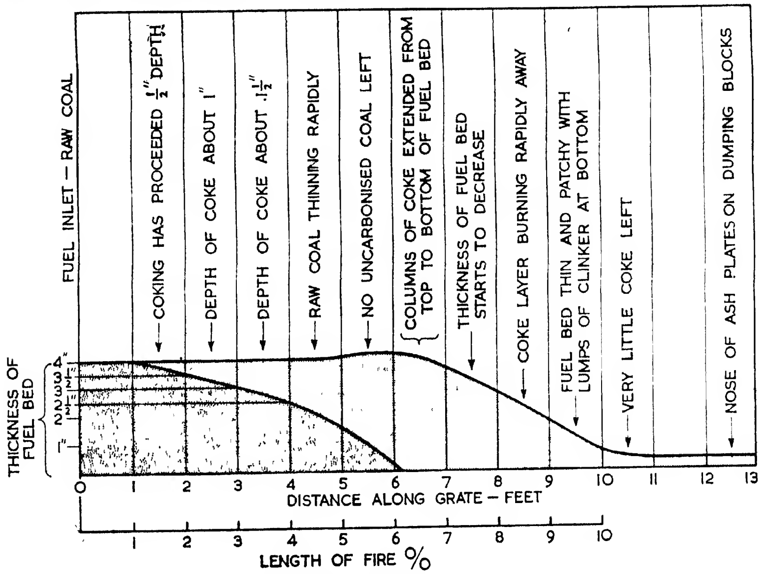


FIG. 9.—COMBUSTION ON A CHAIN-GRATE STOKER

(Reproduced by courtesy from "The Efficient Use of Fuel")

fires require further amount of draught. Later the front air doors can be closed and the forced-draught fan and the secondary air fan placed in operation, due regard being taken of the damper settings.

If the fire remains on the grate for any lengthy period and the grates are not revolving, the danger lies in overheating that portion of the grate. It is wise, therefore, to turn the grate occasionally. Provided the fires are burning near the ignition point, and not likely to prolong distillation and ignition, the speed may be increased until the firegrates are covered.

Before ignition can take place within the combustion chamber with the guillotine doors closed and the coal mechanically fed, the refractory arches must attain a temperature sufficient to promote ignition, with the forced, induced, and secondary air fans in operation.

Controlling Fire Thickness and Grate Speed

By watching through an observation door, it is possible to control the fire thickness and grate speed, as the distance is noted where ignition takes place from the fire side of the guillotine door. Should this distance be more than 6 in., reducing the grate speed may bring the firing nearer to the guillotine door.

A second observation door is situated midway between the beginning and end of the chain grate, and the traverse of the burning mass towards the end can be judged, in so far as the concern is the rate of burning, the amount of

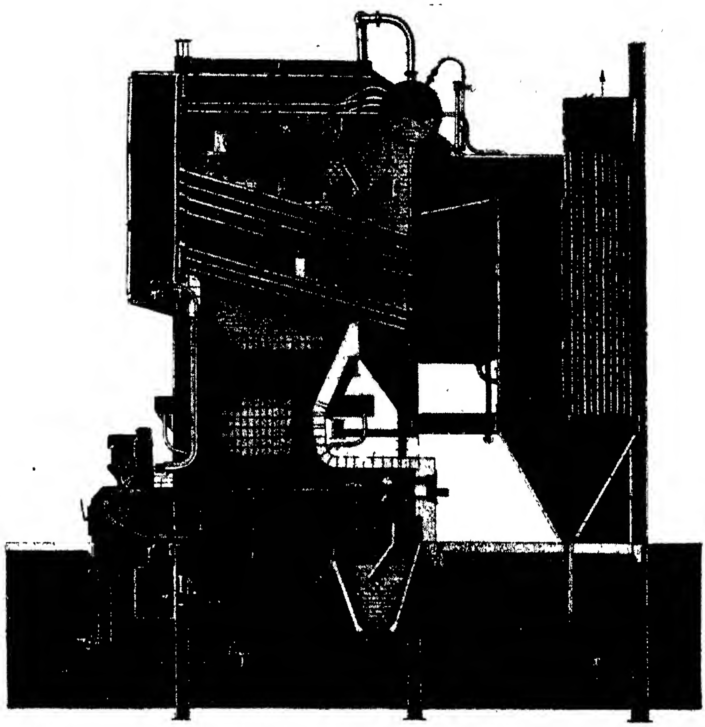
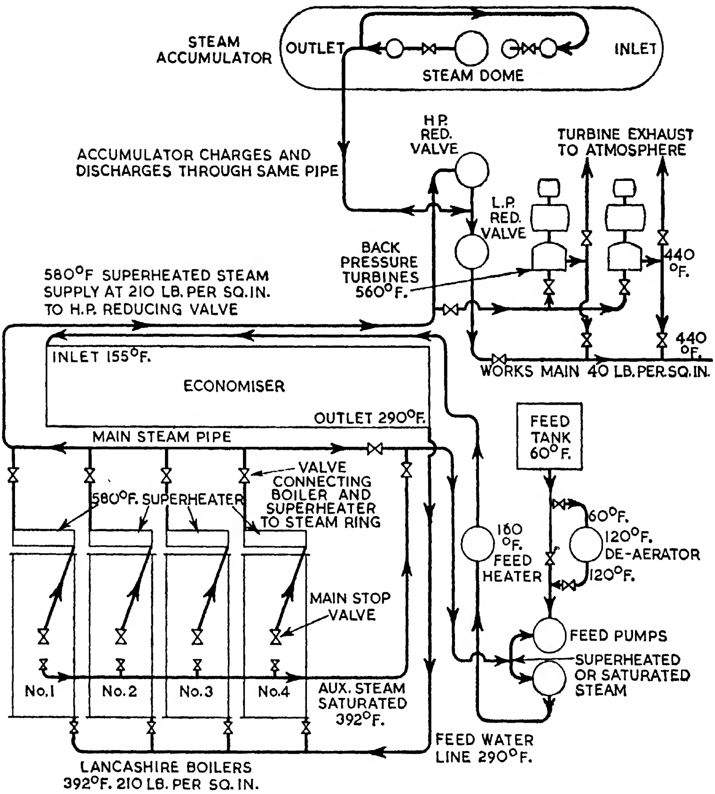


FIG. 10.—C.T.M.-TYPE BABCOCK & WILCOX BOILER

Incorporating forced draught, travelling grate, Bailey furnace, superposed self-cleaning superheater, steel tube economiser, and tubular air heater.

draught, and the behaviour of the coal. Finally, from the rear of the boiler observation doors can be discerned the distance from the dumping blocks where the ashes are likely to form and the coal to be completely consumed. This usually is 18 in. from the dumping blocks. The danger lies in a fast speed, thick fires, and insufficient draught, allowing the coal to pass over the dumping blocks incompletely consumed.

Immediately all air valves are closed and the fires become effective in raising steam, the boiler can slowly be brought to working pressure. It will be noticed, as is also the case in raising steam in the Lancashire-type boilers, that the pressure increases slowly from zero on the pressure gauge, but above 50 lb. per square inch the generation increases as the fires become effective and the unit absorbs heat.



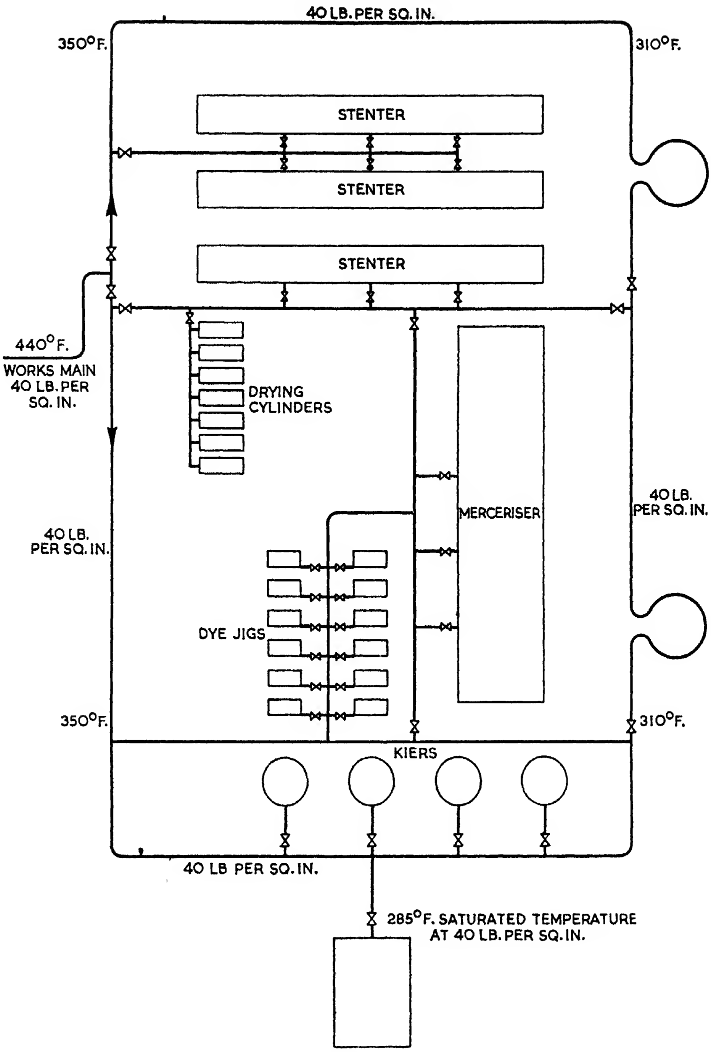
oiler pressure 210 p.s.i. (or lb./in.²) = 392° F. saturated temperature.
uperheat 580° F.—392° F. — 188° F. of superheat.
ize of boilers 30 ft. × 8½ ft. ≒ 10,000 lb. per hour evaporation per boiler.
ccumulator pressure range — 175 p.s.i. charged, 40 p.s.i. discharged. This means 175—40 = 135 p.s.i. drop.

FIG. 11.—COMBINED SYSTEM OF POWER AND PROCESS STEAM WITH STEAM ACCUMULATOR. THE PROCESS STEAM LAYOUT FOR DYE WORKS IS SHOWN OPPOSITE

High-pressure reducing valve decreases boiler pressure to accumulator working pressure between 175 and 40 p.s.i.).

Low-pressure reducing valve cuts down pressure from above 40 p.s.i. to 40 p.s.i. in the works steam main.

The turbines operate at 210 lb. per square inch against a back pressure of 40 p.s.i.



ILLUSTRATIVE DYE WORKS LAYOUT
AND APPROXIMATE STEAM TEMPERATURES

FIG. 11 (*continued*)

Thus, to prevent excess pressure, and the safety valves relieving before working pressure is reached, if this is 210 lb. per square inch, the rate of generation can be controlled by placing the secondary air fan and forced-draught fans out of operation, and opening the front air doors, thus operating an induced draught alone until the boiler is ready for connecting to the main steam line throughout the factory; this action must be adjudged early during the operations. It is assumed, of course, that precautions have been taken regarding valves, and that the water gauge was tested before placing the boiler in service.

Preparing Auxiliary Plant for Operation

The importance of preparing the auxiliary plant during steam-raising operations cannot be overlooked, insomuch that both matters should receive attention during the same period. It is of primary importance that the water supply to the boilers should be available immediately steam is admitted to the main steam line. It is equally important that the feed pump should be in action at this time. This involves other auxiliaries. Therefore, they should receive heat sufficiently in time to ensure free drainage, free operation, and freedom from defects. When approximately 70 lb. per square inch is visible on the steam gauge, steam can be admitted to the auxiliary steam line after ensuring that all drains are open to prevent the action of water hammer. This operation is performed without undue haste. Thus, heat is received by the feed pump and other branch steam lines.

Testing the feed pump during the early stages of plant preparation ensures that no difficulty will be experienced when it is required for duty. This involves the economiser, the feed heater, and the de-aerator, if installed.

Obviously there should be a clear passage for the water to be pumped into the boiler plant. A measure of safety to be taken is to commence opening all the essential valves of the water line, starting with the feed valves on the boilers and subsequent valves on the discharging side of the feed pump, such as the economiser outlet and intervening valves, the economiser inlet and feed-pump outlet valves, and all valves on the suction side of the pump. The steam exhaust to atmosphere is opened, and gradually the control steam valves and drains are closed. Assuming the plant consists of Lancashire boilers, the waste gases will be passing direct to the chimney, until the feed-water temperature is sufficiently high (above 100° F.) to discharge the flue gases through the economiser.

When starting the pump, and in case of air-lock, valves suitably placed on the feed-water line are opened to expel air from the system, and the pump is allowed slowly to commence pumping. It will be noticed that indication of water passing into the boiler is produced by a rise in water level, and due to the pulsating action of the pump, the water in the gauge glass appears to rise and fall whilst increasing. A further precaution is to secure a sufficient supply of water in the feed-water tank to be used when the plant is in full operation.

Placing Economiser in Operation

It is not essential to place the economiser in operation until the plant has been fully prepared to take the steam load, as the water in the economiser may become unduly heated above the normal outlet temperature, which is usually around 300–320° F., due to insufficient supply of water flowing through. Therefore, the flue gases need not be transferred from the chimney base to the economiser circuit until the plant is supplying steam to the factory steam mains.

When it becomes necessary to bring the economiser into operation, this is carried out by opening the main damper exit from the economiser, then the economiser outlet damper, and later the inlet damper when the feed water is circulating through at 100° F. or above. If an induction fan is installed, the cut-out damper is situated probably between the inlet and outlet fan dampers. This should be closed after the fan is started, and both fan dampers open.

Preparing the Plant for Load

As the steam pressure at this stage should be increasing slowly towards 150 lb. per square inch, steam may be gradually admitted to the main factory steam line. The feed pump is in operation at this period to increase the feed-water temperature, and the exhaust from the pump is directed into the feed heater, thus raising the temperature to 150° or 160° F., continuing in this manner until nearing working pressure.

As the working pressure is approached, it becomes necessary to remove from the superheater tubes any condensate which may have collected during the steam-raising period. It is good practice to open the drains early during steam raising, not only to remove condensate, but, if the drains discharge into the feed-water tank, this heats the feed water and also minimises the amount of make-up water required for the boiler-feed supply. If proper drainage is not provided, water may pass into the engine or turbine with disastrous results.

Each boiler can now be separately connected to the main steam pipeline when within a few pounds of the working pressure, the gases transferred from the chimney base to the economiser, the fires conditioned, dampers regulated, and full operational duties carried out.

When the factory steam mains are heating, it is advisable to examine all steam traps and open the trap by-pass valves, thus ensuring that air is expelled and surplus condensate water. During this operation, as steam proceeds through the mains from the boilers, the boiler-water levels decrease. It is during this period that the feed pump should receive the necessary attention to ensure sufficient water being maintained in the boilers.

When pressure is commencing to rise, all traps on the steam mains should operate on direct steam and discharge the condensate into the return pipe to the feed tank or otherwise. If this safeguard does not receive attention and turbines are used for electrical generation, the danger lies in condensate passing to the vanes. Also, in the case of reciprocating engines, severe shocks and possible damage may ensue, due to the small amount of clearance allowed between the pistons and the cylinder ends.

Loading Conditions

When the plant is in full operation, that is, each individual boiler is steaming at working pressure, connected to the main steam line, with mechanical stokers in operation, the induced draught fan, and the feed pump supplying water to the boiler through the economiser, the feed heater and the de-aerator working, the temperature averages from feed tank to the main steam pipe are as follows:

Feed-water tank	60° F., average.
De-aerator outlet	120° F., average.
Feed-heater outlet	160° F., average.
Economiser outlet	300° F., average.
Temperature of saturated steam	392° F., average.
Temperature of superheated steam	580° F., average.
Working steam pressure	210 lb. per square inch.

Flue Temperatures

Side-flue exits	700–800° F.
Economiser inlet	600–650° F.
Economiser outlet	400–450° F.
Chimney base	300–350° F.
Draught at side-flue exits	0·25 w.g.
Draught at chimney base	1·05 w.g.

Operating Efficiency

Should the plant consist of four Lancashire boilers rated at 10,000 lb. of steam per hour, this would be 40,000 lb. of steam at full capacity and evaporating at 7 lb. of steam per pound of coal. In this case the consumption would be 40,000 ÷ 7 = 5,714·3 lb. coal, and coal consumed per boiler hour 5,714 ÷ 4 = 1,428·6 lb.

If the grate area is 40 sq. ft., then coal consumed per square foot

$$= \frac{1428 \cdot 5}{40} = 35 \cdot 7.$$

If the total heat in the steam is found from the Steam Tables to be 1,309 B.Th.U.s when superheated to 580° F., and the calorific value of the coal 11,000 B.Th.U.s per pound, then the overall efficiency of boiler, economiser, and superheater is

$$\frac{7 \times (1309 - 160 - 32)}{11{,}000} \times 100$$

$$= 75 \cdot 1 \text{ per cent.}$$

The total heat usefully employed is therefore 75·1 per cent. of 11,000 B.Th.U.s = 8,261 B.Th.U.s.

The amount of heat wasted or lost = 2,739 B.Th.U.s or 24·9 per cent.

If four boilers are in operation 12 hours per day during 5 days per week

$$= 4 \times 12 \times 5 = 240 \text{ hours.}$$

Also, two boilers operating an additional 12 hours each on night shift (6 p.m. to 6 a.m.)

$$= 2 \times 12 \times 5 = 120 \text{ hours.}$$

Therefore, the total hours steamed per week will be

$$240 + 120 = 360 \text{ hours.}$$

At 1,428·5 lb. of coal per boiler hour, this is

$$\frac{1428{\cdot}5 \times 360}{2240}$$

in tons per week = 229·58.

At 50*s*. per ton the coal costs

$$= \frac{229{\cdot}58 \times 50}{20} = £573{\cdot}95.$$

If each boiler evaporates 10,000 lb. of steam per hour, pounds of water evaporated weekly

$$= 360 \times 10{,}000 = 3{,}600{,}000 \text{ lb.}$$

The cost of evaporation per 1,000 lb.

$$= \frac{\text{Cost of coal in pence}}{\text{Total water evaporated in 1000 lb.}}$$

$$= \frac{573{\cdot}95 \times 240}{3600} = 38{\cdot}26d. \text{ per 1,000 lb.}$$

It is therefore important that the cost per 1,000 lb. of water evaporated should be at the minimum. This can be achieved by selecting suitable coal, not only for the higher calorific value, but also with due regard to suitability. By introducing a blended mixture, maintenance costs can be reduced, and operating efficiency maintained.

W. L.

YARROW LAND WATER-TUBE BOILERS

AFTER a new boiler has been erected by the manufacturers and before completing the lagging, the following operations and inspections should be carried out under the supervision of a responsible engineer:

(1) Flues, ducts, boiler passes, and fan casings should be cleaned and inspected.

(2) Dampers must be checked to ensure that they move freely from the "open" to "shut" positions, and that each index is correct.

(3) The inside of the boiler drums should be wiped clean, wire brushed, and swept out, and the grease used when expanding the tube ends removed.

(4) The insides of all tubes should also be wire brushed and the internal gear in the steam drum replaced.

After these preliminaries, the inspection or searches described next must be carried out on every tube in the boiler, to make sure that there are no obstructions:

(1) Straight tubes should be sighted from the steam drum, using a light held at the bottom of the tube in the water drum.

(2) Bent tubes may be searched by passing through them a suitable size of ball on a string.

(3) Water wall tubes, uprisers, and down comers must be searched in a similar manner.

(4) The superheater tubes should also be inspected by blowing a wooden ball through them, and when the search is completed the superheater internal gear should be replaced and carefully jointed.

As soon as the engineer is satisfied that no foreign material is left inside the boiler and superheater, the manhole doors may be jointed in place. A certificate to the effect that the boiler has been examined and is free from obstruction should then be signed and filed at the boiler manufacturers' head office.

When the manhole doors have been fitted, the fans should be checked for direction of rotation and run for several hours with the dampers very slightly open. Where bearings are water cooled, the water flow should be observed to ensure that it is sufficient. If a mechanical stoker is fitted, it should be run for twenty-four hours at varying speeds. The feed pumps must also be given a trial run and the water supply checked.

Warming up and Boiling out

The next procedure is to fill the boiler with feed water until the water level shows one-quarter in the gauge glass. Washing soda (7 lb. for every

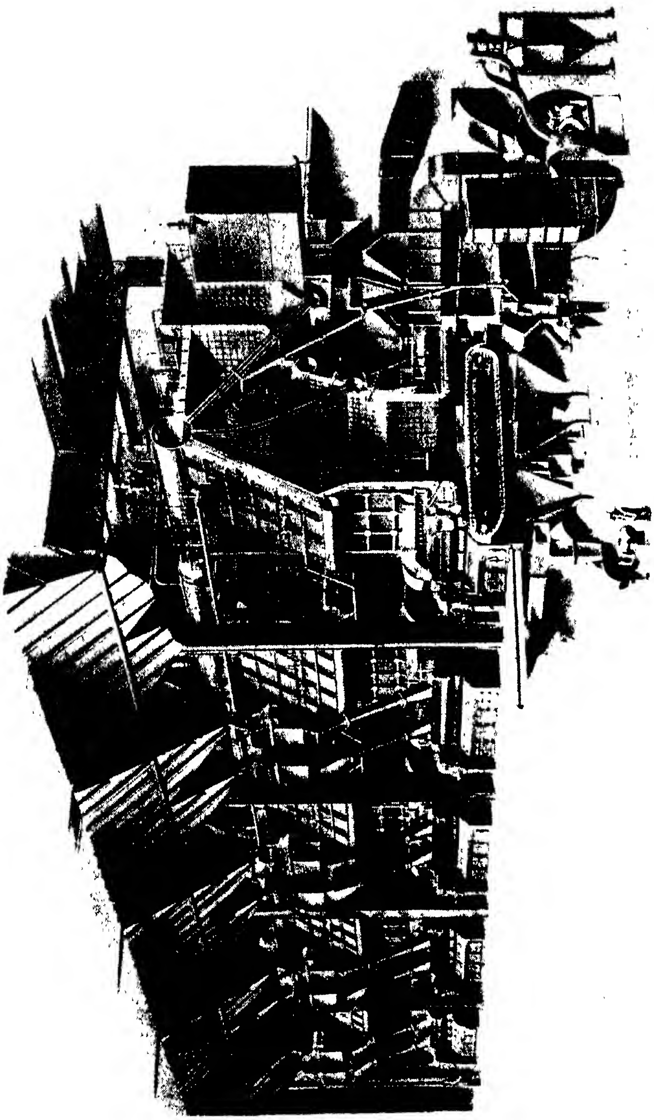


FIG. 1.—FIVE YARROW BOILERS AT CASTLE MEADS POWER STATION, GLOUCESTER
Evaporation: 100,000 lb. per hour M.C.R. per boiler. Steam pressure: 425 lb. per square inch. Final steam temperature: 825° F.

ton of water in the boiler) should be dissolved in water and passed into the boiler.

Then the safety-valve rings must be removed. The blow-down valves on the water drums and the water-circulating valve between steam drum and the blow-down pipes should be fully opened and the blow-down water master valve kept shut. If the boiler is fitted with warming-up pipes and fittings, steam, when available from an external source, should be used to assist in circulating the water from water drums to steam drum until all water in the boiler is heated to about 200° F. The fires may then be lit. If no warming-up system is installed, the fires should be lit when the boiler is cold, but the blow-down valve should still be operated as stated above. This procedure avoids pockets of cold water remaining in the bottom of the water drums with risk of drum distortion. It is a good practice to heat the boiler slowly.

Slow fires should be maintained steadily for about one week to dry the brick settings. If the boiler is fitted with a mechanical stoker, this should be moved forward a few feet every four hours and the fire drawn towards the front of the boiler. This will avoid local overheating of the grate. During the drying-out period, the air valve at the top of the saturated steampipe and the superheater drains should be kept partly open and a pressure of 15 lb. per square inch in the boiler must not be exceeded.

On completion of the drying-out period, the fires should be increased and the pressure raised to about 50 lb. per square inch for a further eight hours, with the air cock and superheater drains still kept partly open. This boiling out is intended to remove all traces of grease from inside the boiler. During boiling out, blow the boiler down by half the gauge glass every hour by easing the master blow-down valve. Loss of water due to evaporation and blowing down should be made up by feeding more water through the economiser into the boiler.

Lagging

During the drying-out and boiling-out period, the lagging should be completed on boiler drums and pipes.

Washing Out

After boiling out, allow the fires to die out and the boiler to cool naturally. When cold, and with no pressure in the boiler, open the air valve fully to kill any vacuum. The boiler should then be emptied, the manhole doors removed, and the boiler ventilated. No naked lights must be allowed near manholes or inside the drums until ventilation is complete; otherwise there is a risk of explosion. The inside of drums and tubes must then be thoroughly washed down with clean water from a hose. If, on examination, the interior is not thoroughly clean and free from grease, boiling out should be repeated.

Care must be taken to ensure that no foreign matter gets into the boiler, and if this is in doubt, a new search of the tubes should be made.

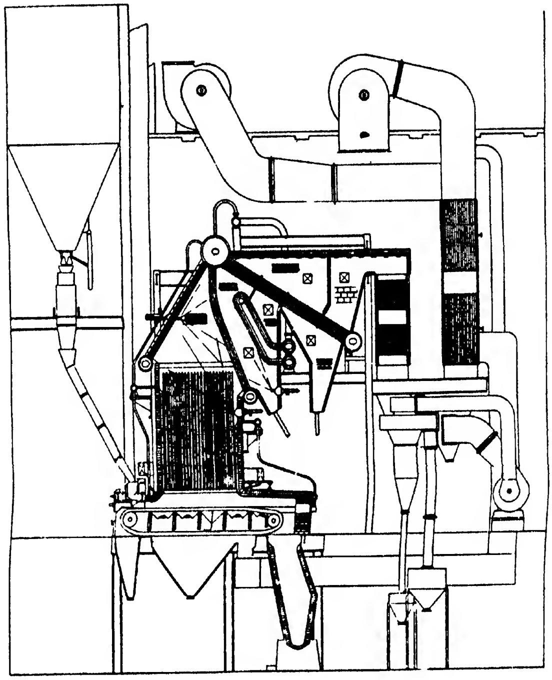


FIG. 2. — VERTICAL ARRANGEMENT OF BOILER

Evaporation: M.C.R. 210,000 lb. per hour

The boiler auxiliaries can be arranged to suit any layout, as shown here and in Fig. 3.

On completion of the washing down, the boiler may be closed up and filled to working level with feed water.

Setting High- and Low-water Alarms

The alarms must now be tested by raising and lowering the water in the steam drum to the levels at which they should operate. For this the hand-feed valve should be used with the automatic-feed regulator isolated. Internal-type alarms are usually tested during the drying out and boiling out period, so that adjustments can be made after the boiler is emptied. External alarms are generally adjusted at a later stage.

Non-steaming Economiser Relief Valves

The economiser, if of steel, should be subjected to full no-load pressure from the feed pumps to see that the relief valves do not lift. Examine these valves to see that they are free and there are no gags in place. With cast-iron economisers, the specified setting of the relief valve is checked by admitting water from the feed range, the valve between economiser and boiler being shut.

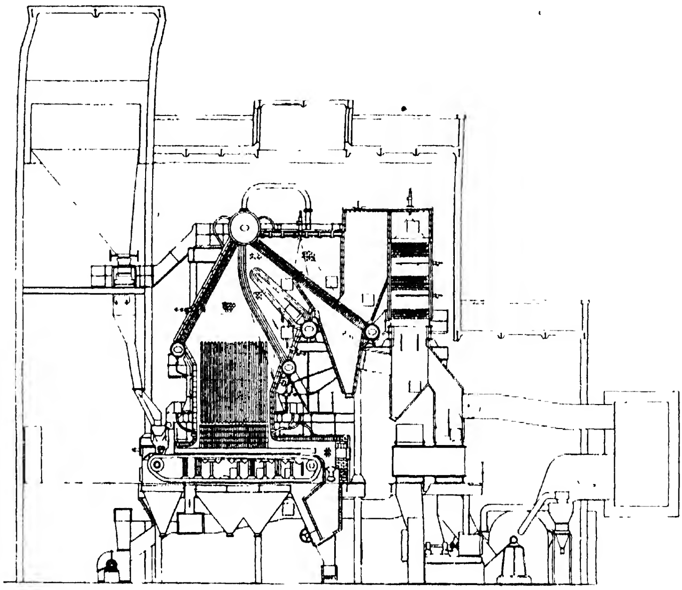


FIG. 3.—HORIZONTAL ARRANGEMENT OF BOILER EVAPORATION: M.C.R. 100,000 LB. PER HOUR

First bringing Boiler to Working Pressure

Proceed by partly opening the air valve on the saturated steampipe and the superheater drain valves. Then open fully the blow-down valves on the water drums and shut the master blow-down valve. The integral blow-down pipes act as connections between water drums, thus avoiding temperature difference, as referred to under "Warming up and Boiling out."

The water in the boiler should now be warmed up by steam from an external source whenever available. Fires should then be lit and kept low, so that the water is heated slowly and evenly.

Adjust the fires to raise the pressure, and when this reaches 10 lb. per square inch, close the air valve, but keep the superheater drain valves slightly open to prevent overheating of the tubes and to prevent accumulation of water.

Slowly raise the pressure until normal working pressure is reached in about four to six hours, depending on the size of the boiler. The external source of, steam for heating should be shut off when the pressure in the boiler is approaching that of the external supply.

Setting Boiler and Superheater Safety Valves

When the correct working pressure has been reached, the safety valves are floated in the presence of a responsible official (usually the Insurance Company's

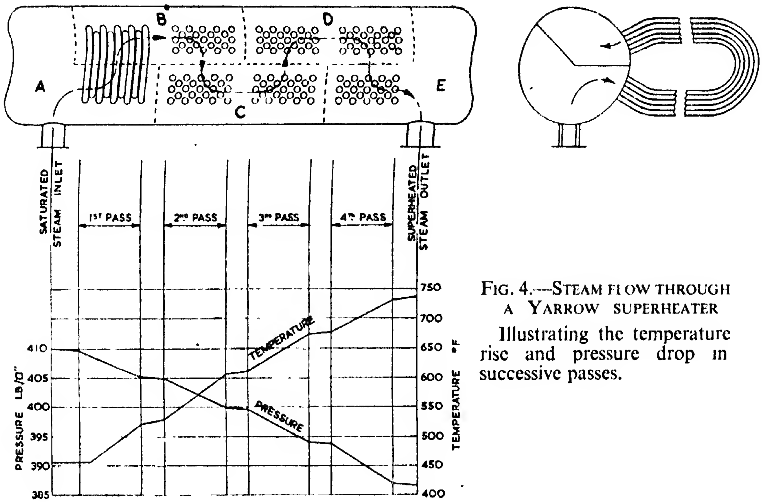


FIG. 4.—STEAM FLOW THROUGH A YARROW SUPERHEATER

Illustrating the temperature rise and pressure drop in successive passes.

surveyor attends). The valve-makers' instructions should be strictly followed. Each valve in turn is tested and adjusted until all valves lift at the specified pressures. The safety-valve rings are adjusted to the correct dimensions, and replaced on the valves.

Superheater safety-valve settings are lower than steam-drum valve settings. This is necessary to maintain steam flow through the superheater tubes in case of sudden shut down. The setting of the respective safety valves must be as specified.

When this work is completed, the valve caps should be put in place and locked, and the key stored by a responsible official. This key should never be available to unauthorised persons. The safety-valve easing gear should then be tried to see that it is operating satisfactorily.

Connecting Boiler to Steam Range

If the steam range is already under pressure, the pressure in the boiler must first be brought to equal that in the range, then open the by-pass valve (when fitted) on the main stop valve slowly, and later the main valve should be opened gradually. When opening to a cold steam range, the stop valve should be slightly eased open to allow the range to warm up slowly and avoid water hammer, and thereafter opened very gradually. The range drain valves must be kept open until the pipes are thoroughly heated and the steam flowing normally. Only after the boiler is delivering steam to the range should the superheater drains and individual blow-down valves be shut.

Soot Blowers

After the boiler is on the range, each soot blower can be tried, but before doing so the soot-blower drain valve should be removed and the pipes blown through atmosphere to expel any dust from the system. After the drain valves are replaced, open the master steam-supply valve slightly and allow time for heating and draining the blower steampipes; this requires about ten minutes. The master valve should then be opened fully and each soot blower operated by turning the handwheel first to the "full on" position and then back to the "steam off" position. This process, taking about one minute, should be repeated three times for each soot blower. When all the blowers have been tried and returned to the "full off" position, the master soot-blower steam valve is shut and the drain valves opened to avoid a vacuum in the pipes drawing in gases which are liable to cause rapid deterioration of the pipes. All furnace blowers must be fully retracted after blowing to avoid damage to the nozzles. It is important that a soot blower should not be used on a cold boiler. All draught gauges must be shut off before soot blowing.

Feed-water Regulator

This should now be tested to see that it operates freely, and is adjusted to maintain the water level at the height in the gauge glass indicated on the boiler mounting drawing.

Instruments

The boiler instruments can now be put into service, and, where necessary, adjusted as makers' instructions.

General

In cases where Yarrow & Co. erect the boiler and set it to work, the above operations are usually supervised by their resident engineer. Thereafter the boiler is available for commercial operation by the purchasers' staff, and the purchaser becomes responsible for complying with Home Office Regulations, the Factories Act, and Electricity Regulations.

Starting up and increasing load should be done gradually. Rapid changes are liable to damage furnace refractory linings. A good guide for moderate- and large-size boilers is to limit the rate of increase in the steam temperature to 100° F. per hour.

OPERATION

The following boiler controls are provided, and are normally operated so that the predetermined steam pressure is maintained:

1. *Thickness of Fuel Bed and Grate Speed.*—Thickness is adjusted by raising or lowering the coal gate at the front of the grate on boilers fitted with chain-grate stokers. Various forms of speed-control mechanism are provided, depending on the type of stoker. The same quantity of fuel can be burnt with a thin fire and high grate speed as with a thick fire and low speed. Judgment

and experience will indicate the combination which gives the best result.

2. *Air Pressure.*—This is adjusted by dampers and/or by varying the speed of the forced-draught fan where speed control is provided.

3. *Air Admission to Fire Bed.*—With mechanical stokers of the compartmented type, control is effected by adjusting the compartment dampers. Dampers in the rear compartments must not be open if there is no live fuel in that zone; otherwise excess air is admitted to the furnace and the boiler efficiency lowered. Rear-compartment dampers should be opened when necessary to prevent unburnt fuel passing over the rear dumps to the ash hopper. Considerable skill and experience are needed to find the combination of compartment damper openings, draught, fuel-bed thickness, and grate speed (see 1) to give the highest efficiency with the coal in use.

4. *Suction Draught.*—This is adjusted by flue dampers and/or by varying the speed of the induced-draught fan where speed control is provided. Where both induced-draught and forced-draught fans are fitted, the suction in the furnace should be maintained between 0·05 in. and 0·15 in. W.G. below atmospheric pressure. Too much suction causes infiltration of air, and too little may result in overheating of furnace fittings and, in extreme cases, the issue of smoke and gases into the boiler-house.

Fuel Firing

(*a*) *Hand-fired Fixed Grates.*—Fire regularly and at short intervals through alternate fire doors. Keep a thin even fuel bed with all corners of the grate covered, including the space at the front between the fire doors. The thickness of the fire depends on the class of fuel and the demand for steam. Too thick a fire bed is uneconomical in fuel consumption. The best results are obtained by an even fire and regulating the draught to give a very light-brown smoke. Firing should be done smartly and the fire doors opened for as short a time as possible to minimise ingress of cold air to the furnace. Coal should not be piled up at the front of the grate and afterwards pushed back. When demand for steam increases, the draught should be increased, and when the demand for steam falls, the draught should be reduced. It is bad practice to control steam pressure by opening fire doors and allowing cold air to enter the furnace. Clean fires, one at a time, and at regular intervals, working rapidly so that the fire door is open no longer than necessary. With some good coals the occasional use of a pricker bar will keep the fires clean for long periods without using a slice.

(*b*) *Mechanical Stokers.*—The aim in operating a mechanical stoker is to secure complete combustion of the fuel on the grate and of the gases in the combustion chamber without permitting excess air to enter the furnace. No hard-and-fast rule can be given for the thickness of fire or for the air pressure for any given steam output, as these depend on the class and size of coal, but the air pressure under the grate should be kept as low as practicable, especially with fine coal. The most efficient setting is a matter for experiment and judgment. It is usual to burn out the fuel on the grate not more than three-quarters down the length of the grate, for the following reasons:

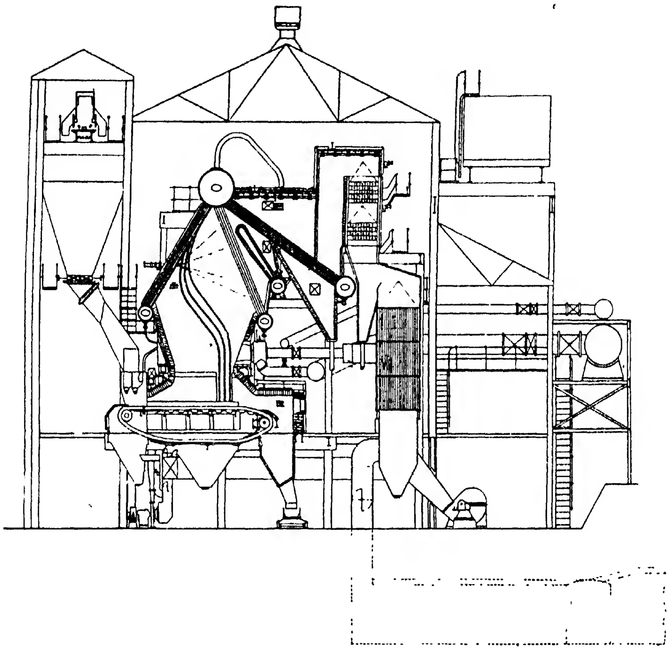


FIG. 5.—YARROW BOILER FOR GAS FIRING

The boiler illustrated is arranged to burn coke-oven gas or blast-furnace gas in conjunction with coke and coal, on a mechanical stoker.

(1) A sudden call for increased steam can then be met by increasing the grate speed without unburnt fuel reaching the dump bars. If holes are burnt in the dump bars, live coals may reach and damage the stoker chain.

(2) If the fire is shorter than three-quarter length, heat under the front arch may be too intense, with consequent damage to the arch blocks.

(3) The length of fire affects superheat, and this factor also may influence adjustment of furnace controls.

(*c*) *Other Fuels.*—When firing pulverised coal, oil, wood, producer gas, blast-furnace gas, coke-oven gas, etc., the instructions issued by the firing equipment makers should be followed.

Control of Air

The total quantity of air admitted to the furnace through the grate and overhead secondary air nozzles is indicated by the CO_2 meter. Too high CO_2 will give incomplete combustion, with smoke or CO at the chimney, and may result in damage to the brickwork of the furnace. Too low a CO_2 reduces the

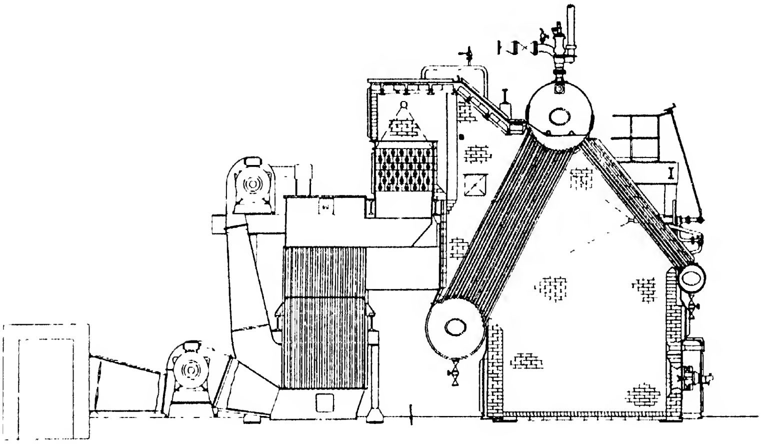


FIG. 6.—YARROW BOILER ARRANGED FOR OIL FIRING

boiler efficiency and increases the fuel consumption. As a general guide, CO is generally of the following order when burning various types of fuel:

	Boiler Exit Per cent.	*Chimney Base Per cent.*
Pulverised fuel	13–14	12–13
Stoker firing (various coals)	12–13	11–12
Oil firing	11–12	10–11

Some boilers are equipped with a steam-flow/air-flow indicator which is set initially by the combustion engineers. Thereafter the fireman adjusts the air supply by observing this indicator instead of a CO_2 meter. Combustion can sometimes be improved by increasing the proportion of secondary air, thus increasing the turbulence in the furnace.

Banked Fires

When a boiler is to be banked, the following procedure should be followed:

(1) Close the dampers (and stop fans if these are individual to the boiler).

(2) Cut off the coal supply.

(3) Stop the grate and push the coal back 3 ft. from the feed door.

(4) Cover the grate in front of the feed door with damp riddlings in order to seal the furnace.

(5) Close the main stop valve.

(6) If the boiler is to be banked for an appreciable length of time, the valve on the steam range should be closed.

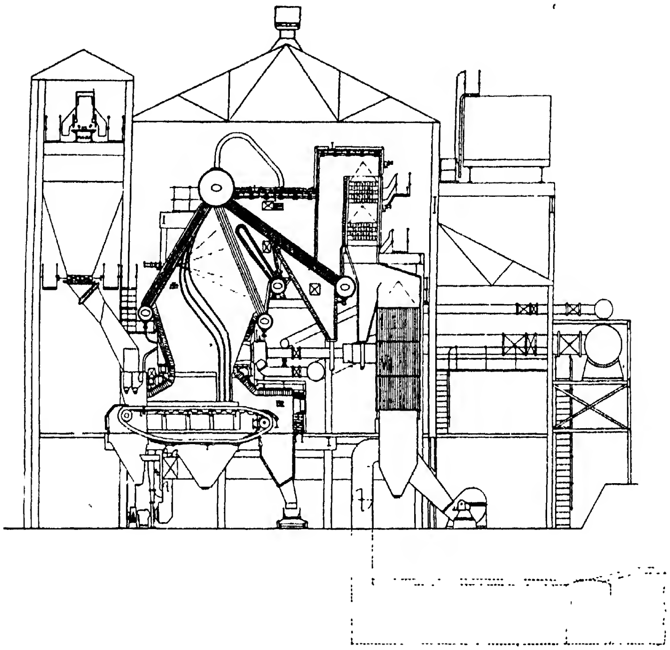


FIG. 5.—YARROW BOILER FOR GAS FIRING

The boiler illustrated is arranged to burn coke-oven gas or blast-furnace gas in conjunction with coke and coal, on a mechanical stoker.

(1) A sudden call for increased steam can then be met by increasing the grate speed without unburnt fuel reaching the dump bars. If holes are burnt in the dump bars, live coals may reach and damage the stoker chain.

(2) If the fire is shorter than three-quarter length, heat under the front arch may be too intense, with consequent damage to the arch blocks.

(3) The length of fire affects superheat, and this factor also may influence adjustment of furnace controls.

(*c*) *Other Fuels.*—When firing pulverised coal, oil, wood, producer gas, blast-furnace gas, coke-oven gas, etc., the instructions issued by the firing equipment makers should be followed.

Control of Air

The total quantity of air admitted to the furnace through the grate and overhead secondary air nozzles is indicated by the CO_2 meter. Too high CO_2 will give incomplete combustion, with smoke or CO at the chimney, and may result in damage to the brickwork of the furnace. Too low a CO_2 reduces the

Emptying Ash Hoppers

Ash hoppers must be discharged at regular intervals to prevent ashes accumulating and covering the stoker ash plates and supports, and so damaging them.

Cleaning Side Walls

Furnace side walls and arches should be kept reasonably clean. Side-wall cleaning, when necessary, should be done while the clinker is hot, as it is then soft and more easily removed by a slicing iron, which should always be kept at the correct angle so as not to damage the brickwork.

Air Heaters

Air entering air heaters should have a minimum temperature of 90° F. In humid countries a higher temperature is necessary to prevent condensation of moisture on the surfaces. Condensation from flue gases results in dilute sulphuric acid attacking metal parts. A good rule is to operate so that the temperature of air entering, added to the temperature of gases leaving the air heater, never falls below a certain figure which, under some circumstances, may be about 330° F. Adjustment of dampers for air and by-passes and for recirculating air enables this condition to be maintained.

Water-cooled Bearings

The nozzles from which the cooling water discharges from the water-cooled bearings should be observed at regular intervals, to ensure that a sufficient but not excessive flow of cooling water is being maintained.

MAINTENANCE WHILST IN SERVICE

Water gauges and gauge cocks must always be kept clean. The water gauge should be blown down at least once every shift, the usual procedure being adopted so as to prove that both steam and water passages are clear and that the true level is being indicated.

Feed Pumps

All standby pumps should be run for at least one hour each week to ensure they can be relied on for immediate service in case of emergency.

Safety Valves

Once a week each safety valve should be eased to ascertain that it is working freely. The setting should be checked if at any time the adjustment is in doubt.

High- and Low-water Alarms

High- and low-water alarms should be tested once a fortnight by raising and lowering the water level in the steam drum to the levels at which the alarm should operate. This test is performed on the hand-feed regulator with the automatic regulator isolated.

Soot Blowers

Soot blowers should be lubricated as makers' instructions.

Bearings on Motors, Fans, Feed Pumps, etc.

All bearings should be oiled or greased regularly as makers' instructions.

Refractories of Furnace Linings and Arches

The greater part of the refractories should last for years without attention, but in the regions subject to more intense heat or scouring action, they may in time fuse, spall or gouge. These parts should be repaired or replaced periodically to avoid accelerated deterioration and consequent damage to other parts by overheating. Metal parts must never be allowed to overheat. If the class of fuel burned is such that repairs are excessive, it may be found helpful to increase the suction in the furnace to 0·2 in. W.G. or to operate at a lower percentage of CO_2, but as either of these involves reduced efficiency and increase in fuel consumption, it should be adopted only as a last resort and then only after expert consultation.

Instruments

All instruments should be checked frequently for accuracy, as incorrect instruments mislead and may result in less efficient operation than if no instruments had been provided.

Casing Leakage

Boiler and economiser casings and flues are unlikely to develop leaks, but occasional observation should be made to avoid infiltration of air into the gas passages.

Hand Lamps

Risk of accident due to faulty hand lamps and leads should be guarded against during boiler repairs and maintenance. Home Office pattern hand lamps should be used, and it is good practice to limit the A.C. supply to 25 volts. Lamp leads should be three-core, one core to earth the frame of the lamp.

OVERHAUL AND MAINTENANCE

Every boiler should be taken out of service at regular intervals for cleaning and repair. It is customary to overhaul each boiler thoroughly once a year and, in addition, to clean and do urgent repairs at other times as found necessary. During the annual overhaul, gas passages, economisers, and air heaters are thoroughly cleaned. Inspecting authorities will examine the condition of the pressure parts and, where required, brickwork will be removed to expose the drums for external inspection. A hydraulic test will also be made and witnessed by the surveyor. Boiler drums and tubes are cleaned internally and repainted in cases where Apexior has been applied previously. If scale forms in the tubes, they should be cleaned before the scale reaches eggshell thickness.

Further Safety Precautions

When a boiler has been taken out of service for overhaul, the following safety precautions must be observed:

All valves capable of being opened to atmosphere should be fully opened, and steps taken to ensure that the boiler is safe for persons to enter. In installations having more than one boiler the steam stop valve, feed valves, blow-off valves, and all other valves and cocks which may be a source of danger must be closed and locked, so that they cannot be opened by accident while overhaul is in progress. An alternative precaution is the removal of a length of pipe between each pipe range and the boiler unit. The latest Factories Act or applicable regulations must always be complied with in regard to precautions against accident. Water should not be thrown on hot flue dust or ashes in a confined space or in any place where this may result in danger.

Tube Replacement

Should it be necessary to replace a boiler tube, cut completely through each end circumferentially, about 1 in. from the drum, and remove the main portion of the tube. The ends left in the tube plates after cleaning off ragged edges can usually be driven into the drums, but if this cannot be done, the bell-mouth inside the drum should be cut off and a slit cut down the tube in the tube hole, great care being taken not to damage the hole in the drum-shell plate. The tube end can then be driven out. Spare tubes are cut to length and ends rounded. When the new tube is in position, the ends are expanded to fit tightly into the tube plate. The ends are then belled with a belling tool until the diameter is not less than $\frac{1}{8}$ in. larger than the plain tube size. After the ends have been belled, they should again be expanded. If the tube is in the middle of a bank, it may be necessary to remove other tubes so that the new tube can be laced or stepped into position by alternately moving the ends into the holes left by the tubes removed for access.

Tubes should have the ends annealed before they are put in place, so that the part to be expanded and belled is in a soft condition.

Precautions after the Overhaul

When a boiler has been cleaned and repaired, and before closing up it must be thoroughly searched as previously described.

The safety valves should be floated and adjusted if necessary at the conclusion of each annual overhaul before the boiler is put into service.

During the overhaul, draft gauge piping should be thoroughly cleaned and all instruments repaired.

Boiler Standing Unused

Useful notes on the preservation of idle boilers are given in B.S. 1170: 1947, clause 18.

A boiler that has been standing unused for a period should be thoroughly examined externally and internally before being put back into service.

We are indebted to Messrs. Yarrow & Co., Ltd., for the information and illustrations used in this article.

YARROW MARINE WATER-TUBE BOILER OPERATION

RAISING STEAM

BEFORE putting a boiler into service after repairs or lying idle for a considerable time, it should be examined to make sure that:

(1) It has been cleared of internal obstructions, such as charcoal burners and lime trays.

(2) All internal fittings and access hole covers are in place and properly secured.

(3) All mountings on which work has been carried out since last steaming are in proper working condition (e.g. gags removed from safety valve after water-pressure test).

(4) All lashed valves are unlashed if work is complete.

(5) All dampers, air doors, fan controls, and safety-valve easing gear are working properly.

(6) Air preheater is by-passed by damper where so fitted.

(7) There is no inflammable material near the boilers; the burner valves are shut, and there are no oil leaks and no oil in the bilges; the sand box, if carried, is full.

(8) The furnace and all air and gas passages are clear and well ventilated; any dampers in the uptakes are in the desired position, and locked if not required for steaming control.

Filling

When the above have been checked, clean distilled (preferably hot) water should be pumped into the boiler, through the running-down valve, if possible, to prevent aeration. Otherwise it should be pumped into the boiler, using both main and auxiliary feed arrangements to check their satisfactory working. The highest air cock on the system and the superheater drains should be open. Stop filling as soon as water is visible at the bottom of the gauge glass. The boiler is then ready for lighting up.

Time Required

If the furnace brickwork is new or has recently undergone repairs, the preliminary heating must be carried out very slowly, using only the smallest burner available as low down and central in the furnace as possible, or a large wood fire may be built inside the furnace while one of the registers on the

FIG. 1.—GENERAL VIEW OF THE BOILER ROOM OF R.M.S. *Caronia*
Yarrow marine water-tube boilers are installed.
(*Stewart Bale, Ltd.*)

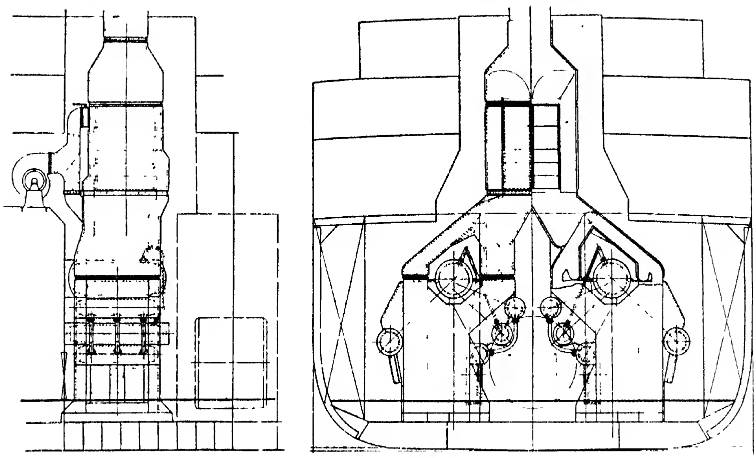


FIG. 2.—ARRANGEMENT TO ILLUSTRATE THE COMPACT LAYOUT POSSIBLE WITH MARINE BOILERS

Note the short overall length of the stokehold and the wide central firing aisle. The boilers are of the five-drum, double-flow, side-fired type, and are fitted with Yarrow superheaters and tubular air preheaters.

boiler front is still open. It may be found necessary to shut off the burner frequently to maintain the required slow rate of heating. When the whole of the brickwork is new, it is recommended that between twenty-four and forty-eight hours be allowed from the time of lighting the fires to connecting the boiler to the main line. If the brickwork is not new, two or three hours should be sufficient, although this may be further reduced if increased brickwork maintenance is acceptable. If the bottoms of the water drums are accessible, these should be felt by hand to ensure that a reasonably uniform temperature is being maintained throughout.

Connecting Up

In raising steam from cold when no steam is available for fans and fuel oil heating and pumping, the auxiliary stop valve should be opened on to the line before heating commences in order that as soon as steam is available from the boiler it can be used for these services. When steam is already available on this line from other sources, these accessories can be supplied without trouble, and the auxiliary stop valve will be kept shut.

When steam issues freely from the air cock, shut it, but leave the superheater drains open and continue heating. Before the boiler pressure reaches the main-line pressure, check the water level, which will then be about full glass, due to the expansion of the water; care must be taken to prevent it passing out of sight when the boiler is connected to the main line; use the blow down as necessary to control this.

Before opening the main stop valve, drain any water out of the main line. When the boiler pressure is equal to the main-line pressure, open the main stop valve very slowly. As soon as the boiler is in use and delivering steam, shut the superheater drains and the preheater by-pass, and follow the routines for normal running.

UNDER WAY

The output of the boiler may be adjusted by varying the number of sprayers in use, the oil pressure, or both. With most oil-fuel systems, the oil pressure may be varied between 75 and 150 lb. per square inch for normal operating conditions. It should not be allowed to fall below 50 nor rise above 175 lb. per square inch. Small variations of output are most conveniently effected by altering the oil pressure; the larger variations by the number of sprayers used. It is recommended that for steady steaming the number of sprayers used should be such that the oil pressure required is between 140 and 150 lb. per square inch, thus allowing for minor variations in output by altering the oil pressure without excessive disturbance of the conditions in the combustion chamber. Generally, a small number of burners at a high pressure for a given output will be more economical and give better combustion than a large number at a lower pressure. For steady steaming the oil pressure must be maintained constant. If the pressure pulsates, check that the oil levels in the fuel-pump discharge air vessels are in sight, but well down in the gauge glass.

Oil Temperature

The oil must be heated to make it sufficiently fluid for efficient spraying. The most suitable temperature at the burners depends on the type of oil, and varies from a maximum of about 220° F. for the heaviest oils down to about 80° F. for shale and Burmah oils. For Texas, Western grade, and similar oils, a temperature of about 160° F. to 180° F. is generally best, and for Persian oil about 130° F. to 150° F. In addition to causing pulsation and unsteady burning, too high a temperature for any given oil renders it liable to "cracking" in the heater, with consequent clogging of the pipes and burners. Too low a temperature, however, may produce late burning in the uptakes, with loss of efficiency, smoking, and injury to air heaters.

Air Supply

The quantity of air required depends upon the amount of fuel being burned and the temperature of the air. For maximum efficiency the supply should be regulated to produce a thin brown haze at the funnel. The light seen through the smoke observation windows will then be clear, and the size of flame from each sprayer of the same pattern will be approximately the same.

Too much air will produce white smoke, and will be indicated by bright flying sparks in the furnace. Too little air will produce black smoke, and may tend to cause late-burning in the uptakes, with overheating of the air

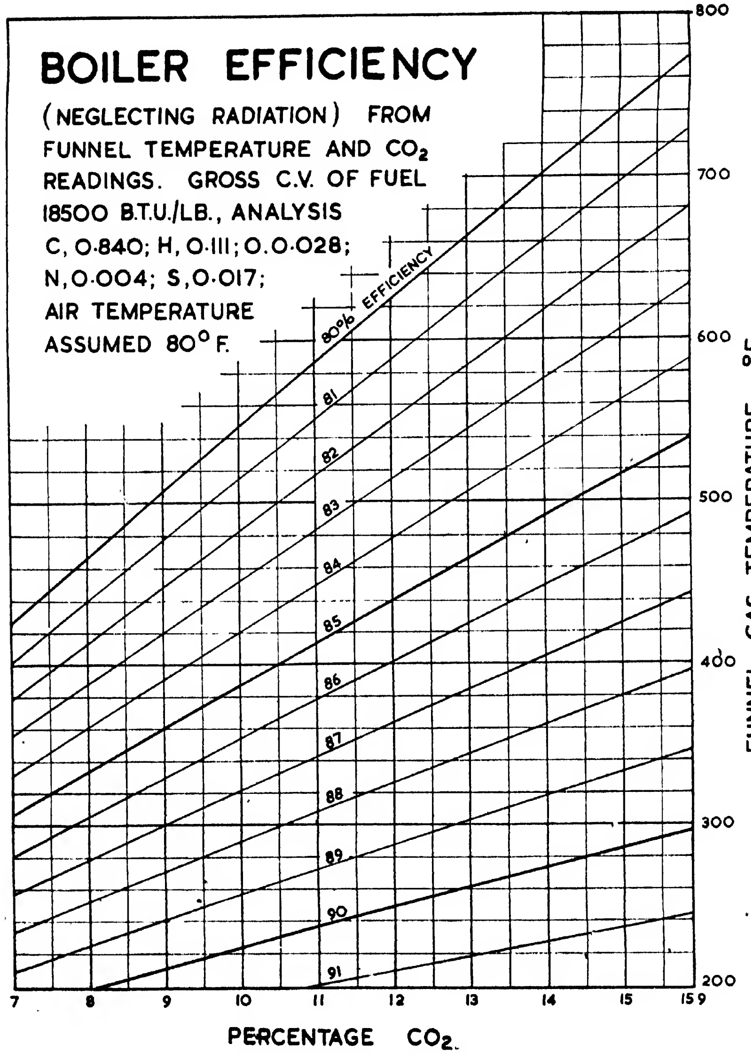


FIG. 3.—BOILER EFFICIENCY CHART

Chart giving the gross efficiency of a boiler from the funnel gas temperature and the percentage volumetric composition of the dry funnel gases.

distributors, preheaters, etc., sooting up of the gas passages and pulsation. The effect of pulsation is for the oil to burn in a series of explosions accompanied by backflash and vibration. If prolonged, it may damage the brickwork. Either of these conditions will reduce the efficiency of the boiler.

Under ordinary circumstances the requisite amount of air can be obtained by regulating the fan, as it is usual to run with the air-box doors fully open. Whenever the fan output is varied, however, watch the flame carefully, as some readjustment of individual air doors may be required. When a burner is shut off, close the corresponding air doors and when easing down or stopping reduce the oil supply prior to the air supply to ensure that any inflammable gas is blown out. If a fan stoppage occurs, shut the oil supply to burners at once.

The use of CO_2, CO, and O_2 indicators, if fitted, simplify the control of combustion. With the combustion controlled as above, the percentage of CO_2 should vary between 11 and 15, depending on the fuel in use; the O_2 should be the minimum (about 3 per cent.) consistent with zero indication of CO. Combustion cannot be improved when any reduction in air supply is accompanied by the formation of CO. If a CO_2 indicator only is fitted, too low a reading indicates too much air. The chart in Fig. 4 shows how the excess air supply may be estimated from the CO_2 or oxygen readings for a typical oil fuel.

Burners

It is of the utmost importance that each burner should be located in its correct position, i.e. set central in and co-axial with its distributor and at the proper distance from the exit end of the distributor, so that with the correct amount of air passing, the spray almost, but not quite, touches the brick quarls or air-tube baffle. On this adjustment depends the efficiency of the whole plant. If the main part of the spray impinges, carbon will form; if, on the other hand, the spray is too far away, the admixture of air and fuel will be imperfect; in either case, unsatisfactory burning will result. Deposits of carbon arising from inefficient combustion may sometimes form on the brickwork at the mouths of the burners and on the floor or side walls of the furnace; these should be removed regularly with the cleaning tool to prevent them becoming extensive and difficult to remove without injury to the brickwork. Steps should be taken at the first possible opportunity to rectify the cause of the trouble. If a burner is dripping, change and repair. One faulty burner may be misleading in producing black smoke without necessarily indicating too little air. There is no harm in the irregular light particles of oil from the spray impinging, as this only causes a feathery deposit which is removed by the slightest touch of a poker and indicates correct setting.

For good burning, it is important that the light outer part of the flame should curl back within the distributor and act regeneratively (i.e. keep the tube hot and dry, not red hot, to vaporise and ignite the incoming oil and heat the combustion air). To induce this action, it may be necessary to shut the air doors momentarily and suddenly release them. If this is not effective and

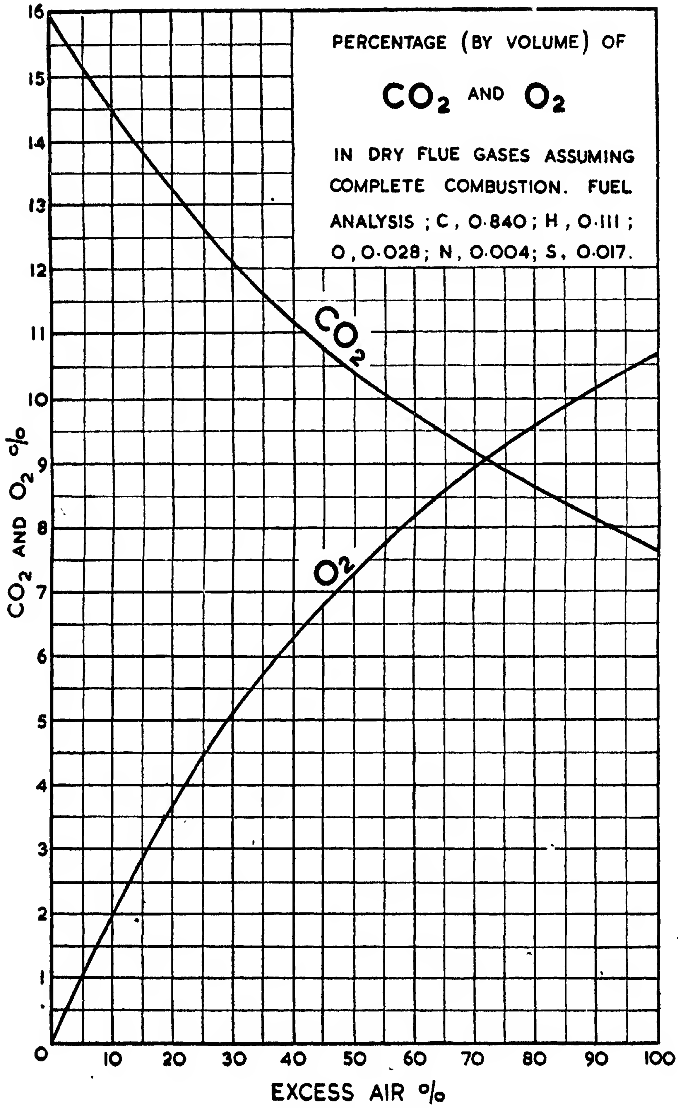


FIG. 4—CHART FOR CALCULATION OF EXCESS AIR

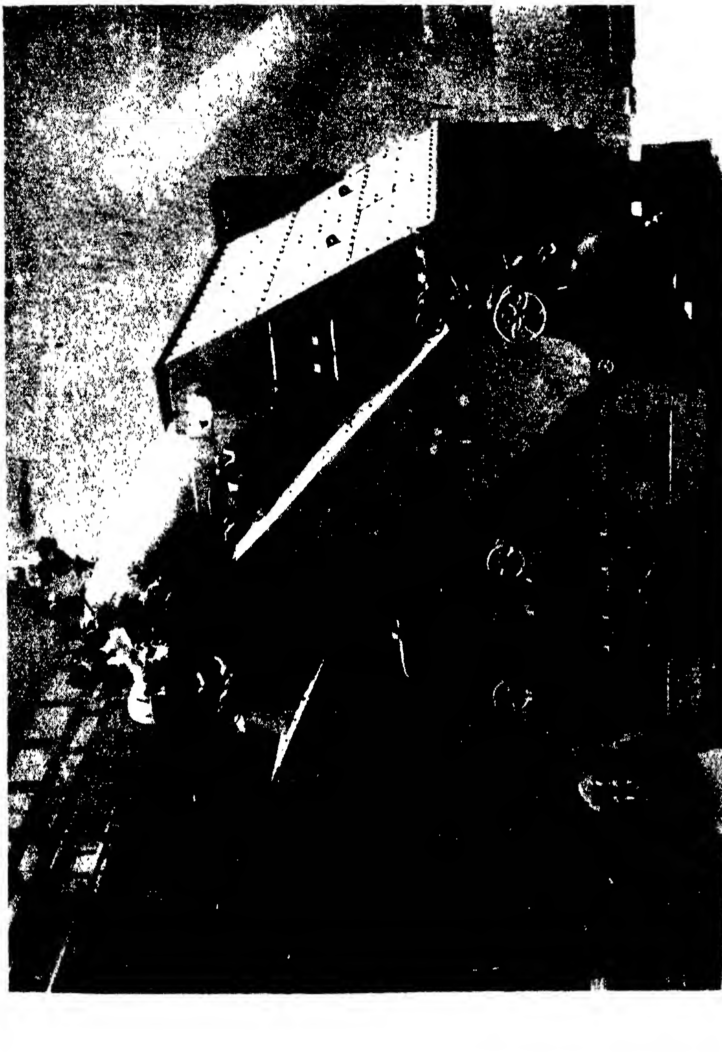


FIG. 5.—TYPICAL YARROW THREE-DRUM BOILER WITH MELESCO SUPERHEATER
As built by R. & W. Hawthorn Leslie for the Furness Withy and Donaldson Lines.

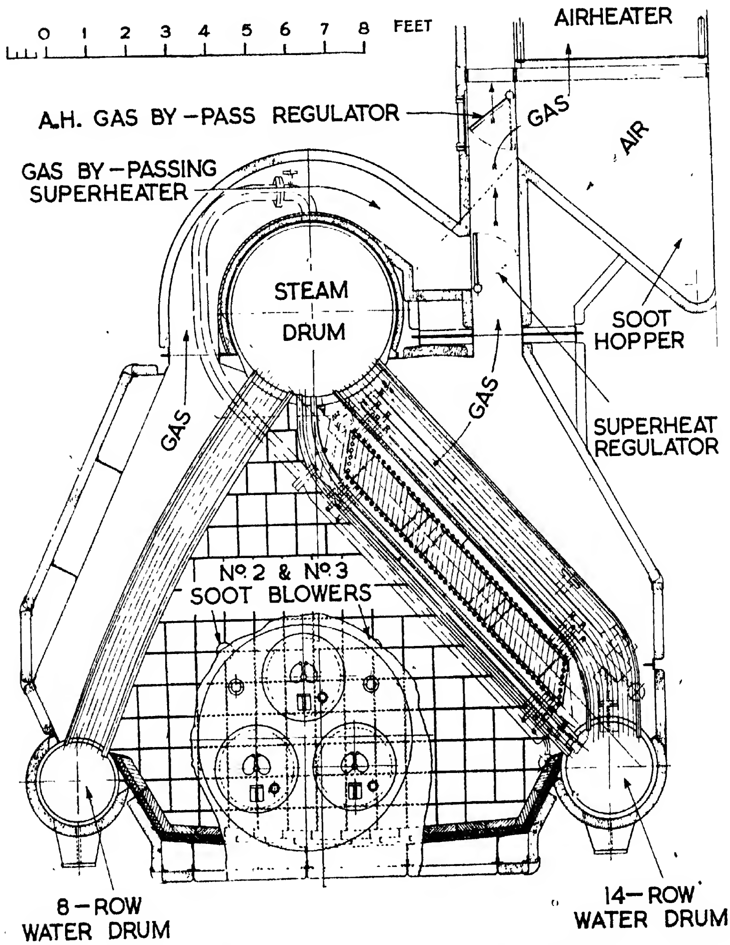


FIG. 6.—DIAGRAMMATIC ARRANGEMENT OF A YARROW THREE-DRUM BOILER FITTED WITH MELESCO SUPERHEATER

The boiler is of the single-flow type, with gas by-pass for superheat control, end firing, an Yarrow vertical tubular air preheater. The normal output is 28,000 lb. per hour, at 455 lb per square inch, and 825° F. from feed water at 300° F.

the combustion tube is wet, the burner may be defective, the oil too low in temperature, or the air supply excessive.

If a burner becomes extinguished, the cause should be traced, and may be due to any of the following:

(1) Air passing over from the air vessel of the oil pump.

(2) Water mixed with the oil from the oil-fuel tanks or leaky heaters.

(3) Choking caused by solid matter, either due to the filters failing to remove foreign matter or to carbonising of the oil.

(4) Too high an oil temperature.

(5) Excessive air supply through distributor.

(6) Blast of steam from sootblowers.

If the trouble is due to water, change the fuel tank and heater in use. As it will be some time before fresh oil reaches the burners, care must be taken to prevent an accumulation of unburnt oil in the furnace. If a burner chokes, remove and clean it thoroughly at once. Burners should be taken to pieces and cleaned periodically, this being carefully carried out to prevent damage to the outlet holes and spindle.

Filters and Strainers

Occasional examination of oil filters and strainers is necessary to ensure their satisfactory operation. Gauges are fitted on each side of the filters so that the pressure difference will indicate how clean they are. The filters are arranged in pairs with isolating valves, so that one can be opened up for cleaning while the other is in use.

Smoke

In general, a fair indication of the combustion efficiency will be given by the colour and density of the smoke and the steadiness of combustion. If the smoke is faint, brown and regular, combustion is probably good. If black or irregular, the normal sources of trouble will be:

(1) Insufficient air.

(2) Oil too cold.

(3) Combustion tubes or furnace dirty or obstructed.

(4) Air doors not open.

(5) Defective burner.

White smoke is generally caused by:

(1) Excessive air supply.

(2) Oil too hot.

(3) Water in the oil.

Manœuvring

When manœuvring, or at any time when rapid changes of load or astern movements may be required, the superheater and the air preheater should be by-passed as far as possible by dampers where so fitted: the former to reduce the steam temperature at the turbines and the latter to prevent excessive

deposits at the outlet end of the air heater due to the lowered gas temperatures. The preheater should be by-passed at light loads also (when operating at funnel temperatures below 250° F.), particularly when the ambient air temperature is low, as in the Arctic regions and in winter in the North Atlantic.

It is also advisable, under these conditions, to keep the standby auxiliaries (oil-fuel pump, oil-heater, feed pump, etc.) idling for use in case of emergency. The water-level should also be watched with even more than usual care, since it is liable to fall out of sight on large reductions of output and rise out of sight on large increases.

Feed-water Supply

When steaming steadily, the feed supply should be maintained as uniform in pressure, temperature, and delivery as possible. Normally, the automatic feed regulators allow this to be done with the feed check valve fully open, but it should also be possible to control the feed by hand without frequent or large movements of the check valve or feed-pump control. If any difficulty is experienced in maintaining the correct water-level, reduce the boiler output and ascertain the cause. When the water-level is out of sight, shut off the burners immediately and test to see whether the gauge glass is empty or full. Most gauge glasses indicate this quite clearly, but in case of doubt blow down the gauge glasses to make sure. Always bring the water-level back into the glass either by auxiliary feeding or by blowing down the boiler as necessary before lighting the burners again.

Boiler-water and Feed-water Purity

The purity of the feed water and boiler water should be given careful attention in view of the higher temperatures and rates of forcing now in use. At boiler pressures above 250 lb. per square inch, it is now recommended that a qualified specialist should be consulted as to the feed treatment required, since each case must be treated separately in the light of different operating conditions and sources of feed water. In general, however, treatment is required if the feed water suffers from any of the following impurities:

(1) Suspended solids, oil, organic matter, etc. These should be removed, first by efficient filtering, and then chemically if extreme purity is required.

(2) Dissolved gases, O_2, CO_2, etc. These are removed mainly by de-aeration and then chemically if necessary.

(3) Permanent hardness salts and other scale-forming salts. These require chemical treatment if present in any appreciable quantity.

(4) Acidity in any form.

Chemical treatment may be employed to counteract these undesirable impurities, but it is much better to prevent their admission to the system by careful distillation of all feed water, and by attention to any sources of pollution, such as leaky condenser tubes, evaporating direct to condenser, disposal of oily drain-water, etc. In addition, certain boiler compounds in the form of paint may be used to prevent any hard scale which does deposit from adhering

to the tubes and drums. These paints are also helpful in preventing conditions arising which might lead to caustic embrittlement.

The troubles associated with the above impurities are priming or foaming, scaling, corrosion, caustic embrittlement, and blowing-down loss. No matter what the treatment may be, it is recommended that a set of routine tests, such as the following, should be adopted to reduce the possibility of dangerous conditions arising in the boiler:

Test	*Water to be Tested*	*Frequency of Tests*	*Requirement*
Silver nitrate	Condensate feed tank	Each watch	No cloud
Boiler density	Boiler (water to be taken from test cock only)	Daily	Less than 100 grains per gallon
Boiler alkalinity	Boiler	Daily	pH between 9·5 and 11
Dissolved oxygen	Feed water	Daily if required	Less than 0·02 c.c. per litre
Oil	Boiler water Drain tank	Regularly	Visual. No sign on gauge glass, etc.

This table may be taken as relating to an installation working at about 450 lb. per square inch. At higher pressures and temperatures, the requirements must be more stringent.

Overload

The Yarrow marine water-tube boiler may safely be forced beyond its normal designed output without ill effects provided combustion remains good, no pulsation is experienced, the water-level is tended with extreme care, and there are no signs of priming. The superheat temperature should tend to rise with output for any given regulator position and feed temperature; any inclination for it to fall should be regarded as a sign of priming. If the purity of the boiler water has not been kept within the prescribed limits, priming will probably be the first indication that the safe limit has been reached; otherwise combustion will be the controlling factor.

Soot Blowers

As soot cannot be entirely prevented even by the most efficient methods of burning, and since the presence of soot on boiler, economiser, and preheater tubes acts as an insulator while the boiler is operating and induces external corrosion while the boiler is lying idle, it is desirable to have some means of keeping the formation of soot down to a minimum. Where soot blowers are fitted, the detailed instructions for the installation should be followed with these points in mind:

(1) Operation of the blowers should be at regular intervals to ensure removal of the soot before it is thick enough to reach its fusing temperature, when no amount of blowing will remove it. Frequency of operation should be found by experience, as it depends on the operating conditions, class of fuel, and efficiency of combustion.

(2) Do not use blowers on a cold boiler.

(3) If wet steam is used, care must be taken to drain away all moisture before blowing.

(4) Isolate draught gauges where fitted to prevent damage.

(5) Alignment of the blowers should be checked periodically to ensure that steam does not impinge on drums and refractory.

(6) If a dust extractor or an induced draught fan is fitted in the uptake, it should be by-passed where possible before blowing. Do not blow soot while the induced draught fan is stopped unless provision has been made to by-pass the funnel gases.

(7) Soot blowers should be used just before shutting down as well as at regular intervals while under way. This should be done on entering harbour while the ship still has way on, to allow the soot to clear the upper decks. They should also be used as soon as possible after lighting up from cold, as soot forms very quickly under these conditions.

(8) See that the blast from the blowers does not extinguish the flame from any oil burner in use.

Observation Fittings

Clean smoke observation windows and mirrors as necessary, but at least once every twenty-four hours. Use them constantly to see that there is no smoke as a general check on the combustion.

Water Gauges

Clean and blow down once each watch. Examine for any traces of oil in the boiler water before blowing down.

Auxiliaries

Run all stand-by pumps occasionally to check their availability for emergencies. Auxiliary feed checks should be worked occasionally (when blowing down is a suitable opportunity).

Leaking Tube

In the unusual event of a boiler tube perforating and leaking into the gas passages while under way, little serious damage is likely to occur if the following action is carried out promptly in the order given:

(1) Shut off the oil supply to the burners.

(2) Increase fan speed.

(3) Shut main stop valve.

(4) Open safety valve.

(5) Maintain feed supply until boiler pressure has gone and boiler has cooled down.

Do not blow the boiler down while there is still any heat in the firebrick.

When the boiler is cooled, the damaged tube should be plugged or cut out and replaced, depending on the urgency with which the boiler is required back in service.

Cleanliness

Great care must always be taken to prevent any accumulation of oil in the air boxes, furnace bottoms, boiler-room floor plates, bilges, etc., and to see that no inflammable material is left lying about. If a leakage from the oil system to the boiler room occurs at any time, shut off the oil supply to that part of the system immediately. Place oiltight trays under all fittings from which liquid fuel may spill when the fitting is opened out. Keep a sandbox in a readily accessible place in the stokehold, and maintain fire extinguishers in efficient condition.

In general, the boiler room should at all times be kept clean and tidy, as this contributes largely to the safety, reduced maintenance, and general efficiency of the boiler installation.

SHUTTING DOWN

In taking a boiler out of service, the following routine should be carried out in the order stated:

(1) Close the oil valves and air flaps to each burner, one by one, until all burners are shut off.

(2) Reduce fan to low speed.

(3) Check that normal water-level is maintained until main stop valve is shut.

(4) Shut main stop valve; the time which must elapse before this is done will depend on the installation, but it should be such that the heat in the brickwork is reduced sufficiently to prevent the boiler pressure rising when the main stop valve is shut. Air doors must not be opened to hasten the cooling of the brickwork.

(5) After cooling for about half an hour, one air door may be opened for about ten minutes to ventilate the furnace and uptakes.

(6) Where the boiler is expected to lie idle for a few days, it is a good practice to pump the boiler as full as possible with hot feed water before it has cooled down. In this case the superheater drains should be closed to allow them to fill up. When the boiler has cooled down, a check should be made on the vents to ensure that the boiler is full of water; if not, the pumping up should be completed.

Note that when using certain grades of oil it may be found necessary to remove all burners immediately on shutting down and clean them by dipping the ends in paraffin.

We are indebted to Messrs. Yarrow & Co., Ltd., for the information and illustrations used in this article.

STEAM PRESSURE REDUCTION AND DESUPERHEATING

FEW works or factories generating steam for power and process purposes utilise all the steam at boiler pressure, and in quite a large proportion steam is required at several different pressures. In some cases it would be difficult to construct process apparatus to work efficiently at boiler pressure, and in others the advantages would be out of proportion to the cost. It is for this reason that reducing valves are to be found in practically all engineering works equipped with steam-generating plants.

A few important and special processes demand a definite temperature, but more often a given quantity of heat in the form of steam at, say, 250° F. is quite as useful and efficient as the same quantity of heat at 300° F. The increase in total heat of steam due to pressure is small, and the latent heat upon which so many processes depend decreases with pressure; therefore the most important factors in process work are usually the volume and the latent heat of the steam. When the necessary volume of steam can be effectively used in a process apparatus, low-pressure steam will generally be quite as useful, and more economical, than steam at a higher pressure. Low-pressure steam is, of course, frequently obtained in a most economical manner by bleeding engines or turbines, or by running the exhausts of auxiliaries to a low-pressure steam main; but as it is not always possible or convenient to obtain the desired quantity of low-pressure steam in this manner, reducing valves must be fitted in the high-pressure steam main.

The Efficient Reducing Valve

An efficient reducing valve should be capable of working with the high-pressure inlet fully opened and should allow steam on the reduction side of the valve to accumulate to the required reduced pressure and then steadily maintain that pressure. Although a number of serious accidents have occurred during recent years following the use of reducing valves, they have not been due to any fault of the valves, but to the absence of safety devices in the process apparatus. It is safe to say that modern reducing valves, manufactured by reputable makers, are perfectly reliable, and, provided suitable types and sizes are selected, and the necessary precautions taken, they will prove efficient and require little attention beyond periodical cleaning and lubrication.

Types of Valve

Reducing valves are constructed according to innumerable designs, and most firms who specialise in this class of work manufacture a range of types and sizes intended to cover all possible requirements. For ordinary industrial use, initial pressures up to 150 lb. per square inch with a fairly constant differential pressure and little or no superheat, the direct-acting diaphragm type of valve having a water seal is generally recommended. A valve of this description is illustrated in Fig. 1. This represents a Hopkinson's "Springus" type valve, in which the diaphragm spring is loaded in accordance with the required reduced pressure. Any fall in this pressure allows the action of the spring to open the valve, thereby admitting high-pressure steam to maintain the reduced pressure. Under actual working conditions there is a continuous flow of steam between the high- and low-pressure sides sufficient to maintain the required low pressure. The self-contained relay type is mostly used for exacting work with high pressures and single-stage reduction, and the balanced full area type of valve is adopted when steam pressure and superheat are very high, with single-stage reduction and fine limits.

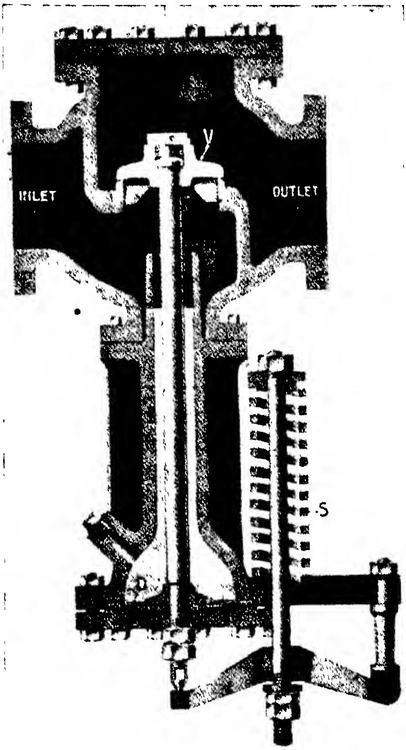


FIG. 1.—DIAPHRAGM-OPERATED REDUCING VALVE

In general industrial process work, where the initial steam pressure is 100 lb. per square inch or less, the old type of directly operated valve, comprising a spring-loaded rubber diaphragm working in conjunction with a single-seated or a double-beat valve, has proved satisfactory and efficient over long periods of work with the minimum attention. This type of reducing valve can only be fitted in a horizontal position, and should be situated at a sufficient distance from the supply or outlet end of the reduced pressure main to allow the greatest possible reserve of low-pressure steam.

The Self-contained Relay Valve

This type is particularly useful when it is necessary to make large reductions from full boiler pressure for heating and process work. The balance chamber must be connected by means of a pipe to a point some 6 ft. or more on the low-pressure side of the reducing valve so that the diaphragm may be influenced by a steady, reduced pressure.

The Full Area Balanced Valve

The full area balanced valve controlled by an external relay regulator has all the control mechanism free from contact with high-pressure steam, and therefore is not affected by high temperature. This class of valve is adopted for the highest pressures and the most exacting type of work.

Selecting a Valve

When a reducing valve is fitted, it is usually expected to function for many years, and, as with all other steam appliances, it is advisable to obtain such valves from firms specialising in boiler accessories. It is false economy to select a particular class of valve simply on account of its cheapness, and it is most important that the makers should be supplied with all the available data.

Valve Size

The quantity of steam that will pass through a reducing valve will depend upon the diameter and length of the high-pressure steam-supply pipe, the type and area of the reducing valve, and the differential pressure. If the inlet pipe is of sufficient diameter to carry the required volume of steam, but the valve too large, it will operate close to its seating with wire-drawing, vibration, and chatter, while if the valve is too small, the steam pressure on the reduction side of the valve will be irregular and difficult to control.

The diameter of the high-pressure steam inlet to a reducing valve is dependent upon the weight and volume of steam to be delivered per hour, the length of the pipe, and the general lay-out of the piping. With steam pipes below 3 in. in diameter and saturated steam, it is usual to base calculations upon a velocity of about 4,500 ft. per minute. If the steam pressure is 100 lb. per square inch and the weight of steam 3,000 lb. per hour, then the sectional area of the pipe will be:

$$\frac{3000 \times 3{\cdot}9 \times 144}{60 \times 4500} = 6{\cdot}2 \text{ sq. in.}$$

and the diameter $\sqrt{(6{\cdot}2 \div 0{\cdot}7854)} = 2{\cdot}8$ in., where 3·9 is the volume of 1 lb. of steam in cubic feet.

A single-seated valve must lift one-quarter of its diameter to give an opening equal to its area and a double-seated valve half that amount. In either case such a lift is beyond the capacity of a diaphragm-operated valve, but in practice, when the reduced pressure is not less than 58 per cent. of the initial pressure,

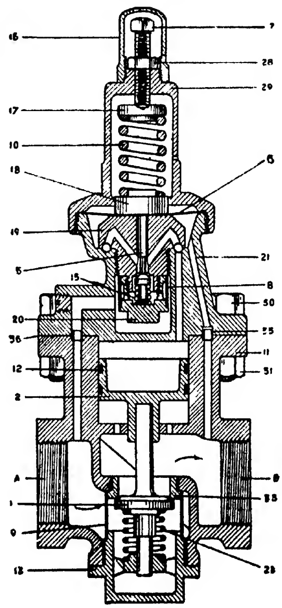


FIG. 2 (*right*).—GUNMETAL REDUCING VALVE

1. Main valve; 2. Piston; 5. Auxiliary valve; 6. Diaphragm; 7. Spring adjusting screw; 8. Auxiliary valve strainer; 9. Main valve strainer; 10. Adjustment spring; 11. Body; 12. Piston rings; 13. Bottom flange (or plug); 15. Auxiliary valve spring; 16. Locking device cap; 17. Upper spring washer; 18. Lower spring washer; 19. Auxiliary valve top plug; 20. Auxiliary valve bottom plug; 21. Auxiliary valve top; 22. Main valve spring; 23. Top flange; 24. Auxiliary valve-top screws; 28. Lock-nut; 29. Spring chamber; 30. Body studs; 31. Body-stud nuts; 32. Piston chamber; 33. Seat ring; 34. Bottom flange bushing; 35. Port bushing.

the increased steam velocity due to the pressure drop and resulting superheat will usually be sufficient to discharge the required quantity of steam when the valve area is equal to the sectional area of the steam inlet pipe.

The capacity of the relay operated type of valve increases with the pressure drop up to 58 per cent., when any decrease in pressure, even below atmospheric pressure, will not increase the steam flow.

Diameter of Discharge Pipe

For satisfactory operation it is essential that the pipe from the reducing valve to the process apparatus should be of sufficient capacity to accommodate the increase in steam volume due to the reduction in pressure. At 20 lb. per square inch the volume of 1 lb. of saturated steam is 12 cub. ft., and to deliver 3,000 lb. of steam per hour at a velocity of 4,500 ft. per minute the diameter of the pipe would require to be:

$$\sqrt{\left(\frac{3000 \times 12 \times 144}{0{\cdot}7854 \times 60 \times 4{,}500}\right)} = 5 \text{ in.}$$

The Foster Reducing Valve

The Foster reducing valve, manufactured by Sir W. H. Bailey & Co., Manchester, is illustrated in Figs. 2 and 3. Fig. 2 is constructed of gunmetal with threaded inlet and outlet, and Fig. 3 of cast iron with flanged connections.

Operation of Valve

Pressure enters at (*A*) and passes through main valve to outlet (*B*). Main valve (1) is controlled by auxiliary valve (5), which is opened by adjusting spring (10) and closed by reduced pressure acting on diaphragm (6). Auxiliary valve is held open by the adjustment spring (10). When supply pressure is admitted to the valve, it passes up through port (*C*) on inlet side of valve to auxiliary valve chamber (*P*); thence to top of operating piston (2).

The Self-contained Relay Valve

This type is particularly useful when it is necessary to make large reductions from full boiler pressure for heating and process work. The balance chamber must be connected by means of a pipe to a point some 6 ft. or more on the low-pressure side of the reducing valve so that the diaphragm may be influenced by a steady, reduced pressure.

The Full Area Balanced Valve

The full area balanced valve controlled by an external relay regulator has all the control mechanism free from contact with high-pressure steam, and therefore is not affected by high temperature. This class of valve is adopted for the highest pressures and the most exacting type of work.

Selecting a Valve

When a reducing valve is fitted, it is usually expected to function for many years, and, as with all other steam appliances, it is advisable to obtain such valves from firms specialising in boiler accessories. It is false economy to select a particular class of valve simply on account of its cheapness, and it is most important that the makers should be supplied with all the available data.

Valve Size

The quantity of steam that will pass through a reducing valve will depend upon the diameter and length of the high-pressure steam-supply pipe, the type and area of the reducing valve, and the differential pressure. If the inlet pipe is of sufficient diameter to carry the required volume of steam, but the valve too large, it will operate close to its seating with wire-drawing, vibration, and chatter, while if the valve is too small, the steam pressure on the reduction side of the valve will be irregular and difficult to control.

The diameter of the high-pressure steam inlet to a reducing valve is dependent upon the weight and volume of steam to be delivered per hour, the length of the pipe, and the general lay-out of the piping. With steam pipes below 3 in. in diameter and saturated steam, it is usual to base calculations upon a velocity of about 4,500 ft. per minute. If the steam pressure is 100 lb. per square inch and the weight of steam 3,000 lb. per hour, then the sectional area of the pipe will be:

$$\frac{3000 \times 3{\cdot}9 \times 144}{60 \times 4500} = 6{\cdot}2 \text{ sq. in.}$$

and the diameter $\sqrt{(6{\cdot}2 \div 0{\cdot}7854)} = 2{\cdot}8$ in., where 3·9 is the volume of 1 lb. of steam in cubic feet.

A single-seated valve must lift one-quarter of its diameter to give an opening equal to its area and a double-seated valve half that amount. In either case such a lift is beyond the capacity of a diaphragm-operated valve, but in practice, when the reduced pressure is not less than 58 per cent. of the initial pressure,

DESUPERHEATERS

The comparative value of saturated and superheated steam for process work is a very debatable point. Theoretically, the latter should be of greater use and the more efficient, but the majority of steam users seem to agree that, taking all the factors into consideration, saturated steam is the more economical and gives the better results. It is probable that, except in a few special cases, superheated steam up to about 60° F. will have little effect upon the results obtained and that more than about 100° F. of superheat is undesirable for process work. For some classes of work, for example vulcanising, superheated steam has a definite advantage.

A large number of processes demand steam. not only at a low pressure, but also within fairly close temperature limits, and when high-temperature superheated steam is obtained by bleeding engines, it is often necessary to reduce its temperature to within the desired limits. For this purpose desuperheaters are used, either in conjunction with reducing valves with pressure reduction, or as a self-contained temperature-reducing unit.

Desuperheaters are frequently fitted in modern steam generating plants in order to control the temperature of superheated steam within the limits of plus or minus 10° F.

Construction

Desuperheaters consist of mild-steel or cast-iron cylindrical vessels or manifolds containing baffles or compartments that ensure the efficient spraying of the steam and are provided with suitable inlet and outlet connections and tappings for the various fittings. Spray water at a pressure sufficient to produce satisfactory atomisation and overcome the internal pressure enters the vessel through one or more spraying nozzles. Hand control of the sprays or automatic operation by means of a thermostat, or a combination of both, may be adopted. A receiver type of desuperheater is shown in Fig. 4. This is constructed by Messrs. Hopkinsons, Ltd., Huddersfield. A sectional view is given in Fig. 5.

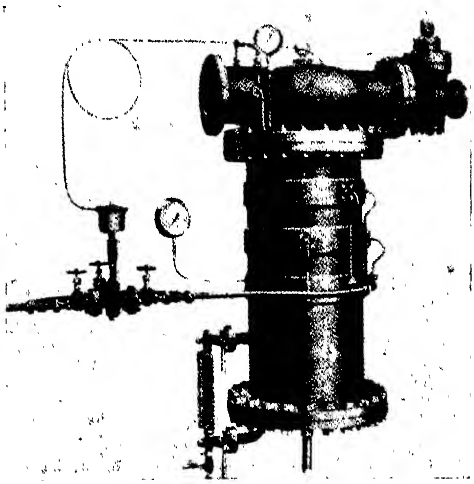


Fig. 4.—Receiver type of desuperheater

This type of desuper-

heater combines steam receiver, separator, and desuperheater chamber in one unit, and when used with a thermostatic water control and a reducing valve, as shown in Fig. 4, makes a simple and compact unit. An electrical regulator can also be used for the water and steam control if preferred. A special feature of the design is that the walls and outlet of the desuperheater are steam jacketed by superheated steam and thus kept at a high temperature. This materially reduces the drainage of any water impinging on the walls, and a further improvement is that any water draining from the walls has to pass through the incoming superheated steam and will be picked up and redistributed into the steam flow.

Another type of superheater consists of a steel vessel in which there is a series of multiple loop steel tube elements through which a certain proportion of steam, heated in the primary superheater, is passed. These tubes are submerged in water at the same level as the water in the boiler; a connecting pipe is led from the steam drum to a point near the base of the desuperheater, whilst a balance pipe also joins the top of the desuperheater and the top of the

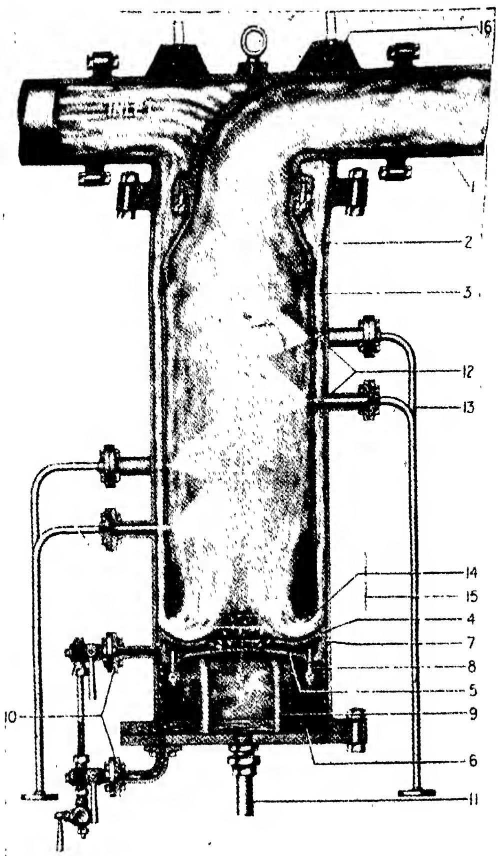


FIG. 5.—SECTIONAL VIEW OF THE HOPKINSON DESUPERHEATER

1. Manifold casting; 2. Outer casing; 3. Inner casing heated by superheated steam; 4. Deflection plate; 5. Baffle plate allowing excess water to drain to sump 6; 6. Water sump; 7. Packing for baffle plate; 8. Spring tension for packing; 9. Feet supporting baffle plate; 10. Water gauge connections; 11. Drain connection; 12. Water spraying nozzles; 13. Water supply to nozzles, supply controlled by thermostat; 14. Drainage edge, water drips into incoming steam; 15. Lagging shield when required; 16. Lugs for supporting desuperheater.

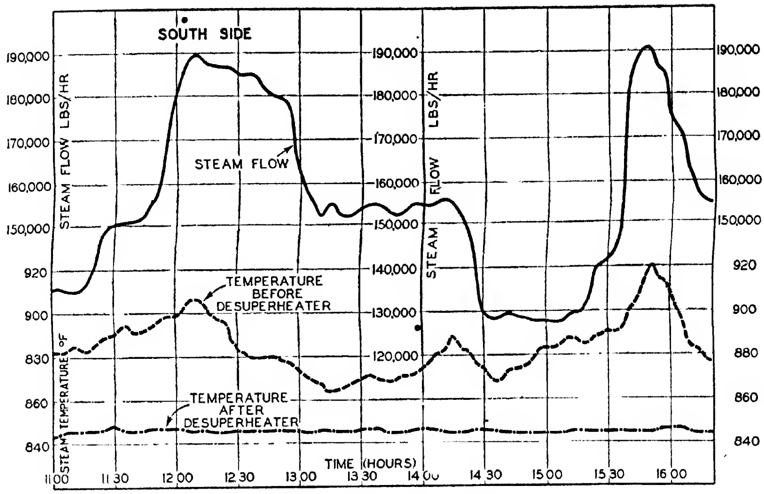


FIG. 6.—CHART SHOWING CONTROL OF TEMPERATURE BY SUPERHEATER

steam drum, through which steam generated in the desuperheater is returned to the boiler. A steam pipe is arranged to by-pass the desuperheater, and butterfly valves control the amount of steam leaving the primary superheater, which must be cooled in the desuperheater before entering the secondary superheater and mixing with the bulk of the steam, in order to maintain the required temperature at the stop valve. The operation of these valves is effected by electric motor through gearboxes and links, and is controlled by a thermostat placed in the steam main leaving the boiler.

Superheating Effect of Reducing Valves

When dry saturated steam is reduced in pressure by passing through a reducing valve, it becomes superheated. If steam at 100 lb. gauge pressure is reduced to 20 lb., its temperature is increased:

$$(1{,}194 - 1{,}169) \div 0{\cdot}56 = 44^\circ \text{ F.}$$

where 1,194 B.Th.U.s is the total heat of steam at 100 lb. from 32° F., 1,169 B.Th.U.s the total heat at 20 lb., and 0·56 the specific heat of steam.

Superheat

The total heat of saturated steam to which superheating has been applied is increased by an amount equal to the degree of superheat multiplied by the specific heat. The total heat of superheated steam can be obtained from a Mollier diagram or from steam tables. When the temperature of saturated steam at 105 lb. gauge pressure and 341° F. is increased by 150° to 491° F., the

total heat becomes 1,269 B.Th.U.s, an increase of 1,269 − 1,190 = 79 B.Th.U.s. To desuperheat such steam it would be necessary to extract 79 B.Th.U.s per lb.

Water for Desuperheating

Assuming that 1 lb. of steam at 205 lb. gauge pressure superheated 150° F. is reduced in pressure to 20 lb. and desuperheated by means of a water spray at 60° F., all the water being converted into steam. Then the total heat of steam per pound entering the desuperheater would be 1,285 B.Th.U.s, and the total heat of the steam leaving the desuperheater 1,169 B.Th.U.s. Therefore, 1,285 − 1,169 = 116 B.Th.U.s must be extracted from the steam and transmitted to the water. If 1,000 lb. of steam per minute enters the desuperheater, it follows that 116,000 B.Th.U.s must be taken up by the water.

To convert 1 lb. of water at 60° F. into steam at 20 lb. gauge pressure requires 1,169 − (60 − 32) = 1,141 B.Th.U.s. Therefore, 101 lb. (i.e. 116,000 ÷ 1,141) of water will be required, and the weight of steam at the reduced pressure will be increased to approximately 1,101 lb.

Automatic Temperature Control

It is difficult to obtain a constant temperature with highly superheated steam when the load is fluctuating widely, and for this reason the desuperheater is used for the purpose of maintaining a fixed temperature, thus enabling a higher temperature to be used with safety at turbine or engine stop valves. The record shown in Fig. 6 indicates the very close control of temperature that can be secured by desuperheating in conjunction with electrical or thermostatic control of the water supply.

Method of Water Supply

When cold water is sprayed into the desuperheater at a pressure below about 150 lb. per square inch injectors can be used, but a feed pump is more satisfactory and reliable, particularly when the spray water is hot.

STEAM INJECTORS, EJECTORS, AND WATER HEATERS

THE modern injector is a most reliable and compact apparatus of reasonable efficiency, and, compared with a pump, the first cost is low, and repairs or renewals are more readily and cheaply carried out. Injectors are largely used for feeding water into locomotive, portable, and all types of stationary boilers, but also for lifting, washing, and spraying. The steam consumption of an injector is greater than that of a boiler feed pump of corresponding capacity, but the amount of steam used by a pump is much greater than is generally realised, particularly after the pump has been in use for a long period. The injector, however, raises the temperature of the feed water and acts as a feed heater, and it is undoubtedly better to use an injector and feed hot water into a boiler than use a feed pump and supply cold water.

The action of an injector in use is due to the momentum imparted to the water by a jet of steam moving at high velocity. All injectors have three principal parts, consisting of a steam cone into which steam at boiler pressure is directed, a combining cone into which the water flows and, meeting with the steam, produces a partial vacuum with a loss of pressure, and a delivery cone where the kinetic energy of the combined water and steam has sufficient pressure to force the stream of water and condensate into the boiler. An escape or overflow allows the injector to start into action by providing an outlet through which steam and water may escape.

Capacity of Injectors

The quantity of water a live steam injector will discharge is dependent upon the pressure and velocity of the steam in the steam cone or nozzle and upon the temperature of the water and the length of suction pipe. The elementary principle is that steam passing through a nozzle is condensed and a partial vacuum created. The mixture of steam and water is then forced with increasing velocity into a convergent nozzle and from thence through a divergent nozzle, where it loses velocity and gains pressure.

To find the diameter of the throat of the delivery cone let d = diameter in millimetres, V = volume of water in gallons per hour, and p = pressure in lb. per square inch steam pressure, then

$$0{\cdot}709 \sqrt{\frac{V}{\sqrt{p}}} = d$$

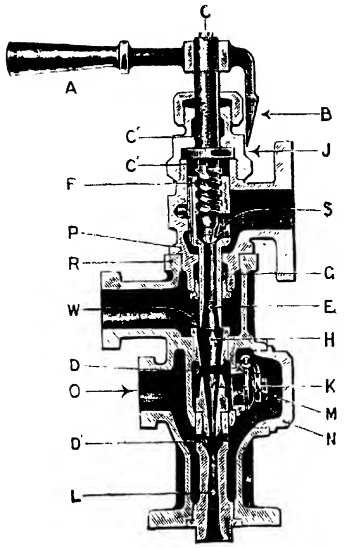


FIG. 1.—A ONE-MOVEMENT INJECTOR FOR PRESSURES UP TO 250 LB. PER SQUARE INCH

(*A*) Operating and regulating lever.
(*B*) Pointer and pressure index for working position.
(*C*) Lubricating hole for spindle.
(*C'C'*) Lubricating holes in spindle.
(*DD'*) Overflows.
(*E*) Steam nozzle.
(*F*) Steam spindle.
(*G*) Steam-nozzle packing.
(*H*) Lifting tube.
(*K*) Combining nozzle.
(*L*) Delivery nozzle and outlet to boiler.
(*M*) Overflow valve.
(*N*) Overflow cap.
(*O*) Overflow outlet.
(*P*) Injector body, top.
(*R*) Injector body, bottom.
(*S*) Steam inlet.
(*W*) Water inlet.

Example.—Let $p = 100$ lb. steam pressure and $V = 500$ gallons per hour, then

$$0{\cdot}709\sqrt{\frac{500}{\sqrt{100}}} = 5 \text{ mm.}$$

Volume of Discharge

The volume in gallons an injector will lift with a suction not exceeding 3 ft. and a temperature not exceeding 60° F. is equal to $1{\cdot}985d^2\sqrt{p}$, where d = diameter of nozzle and p = steam pressure.

Example.—Diameter of nozzle 5 mm., steam pressure 100 lb. Then $1{\cdot}985 \times 5^2 \times \sqrt{100} = 495$ gallons.

The quantity of water lifted is approximately 3 per cent. less for each foot of suction lift above 3 ft. and decreases from about 5 per cent. when the water temperature is 85° F. to about 25 per cent. at a temperature of 135°; with higher temperatures the discharge may be reduced as much as 50 per cent. A combination of high lift and high temperature should be avoided.

Steam Consumption of Injectors

The weight of steam used in working an injector will depend upon its pressure and dryness fraction, and the volume of water discharged will vary from about 1 to 1·8 gallons per lb. of steam, according to the suction height and the temperature of the water lifted. If 1 lb. of steam at 80 lb. pressure lifts 15 lb. of water at 60° F., the temperature of the mixture will be $15 \times (60 - 32) + 1{,}190 \div 16 = 100°$ above 32° or 132° F.

The ratio of water to steam can be found approximately by

$$R = \frac{H - t_2}{t_2 - t_1}$$

where R = ratio of water to steam;
H = total heat of steam;
t_1 = initial water temperature;
t_2 = final water temperature.

Thus, if $H = 1{,}190$ B.Th.U.s, $t_2 = 132°$ F., and $t_1 = 60°$ F., then

$$\frac{1190 - 132}{132 - 60} = 14{\cdot}6 \text{ lb. of water per lb. of steam.}$$

Types of Injectors

A standard type of one-movement injector suitable for pressures of from 25 lb. to 250 lb. per square inch is shown in Fig. 1, with the various components indicated. The injector is manufactured by Messrs. Holden & Brooke, Manchester, in twelve sizes.

The following figures give the lift and the lift temperatures for various pressures:

LIFT TABLE

Steam pressure in lb. per square inch . .	25	40	60	80	100	120	140	160	180	200	225	250
Height of lift in feet . .	3	15	18	20	20	20	20	20	20	20	20	20

LIFT AND PRESSURE TABLE

Steam pressure in lb. per square inch . . .	25		50				100			
Lift in feet	0	3	0	3	6	10	0	3	6	10
Maximum temperature of inlet water °F.	120	115	135	130	120	110	128	123	115	105

Steam pressure in lb. per square inch . .	150				200				250			
Lift in feet . . .	0	3	6	10	0	3	6	10	0	3	6	10
Maximum temperature of inlet water °F.	117	112	107	100	109	104	100	97	83	80	75	70

Fixing the Injector

This type of injector will work in any position, provided the overflow valve is free to fall on to its seat. The two most usual positions are shown in Fig. 2.

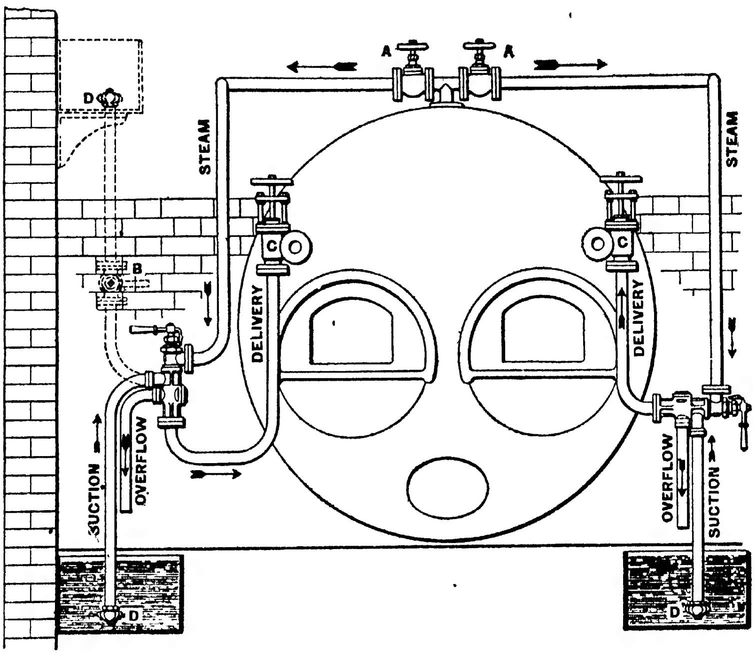


FIG. 2.—ALTERNATIVE METHODS OF FIXING INJECTOR FOR BOILER FEEDING

Tail Pipes.—In the case of injectors with union and not flanged connections, do not use pipe wrenches or dogs on the tail pipes, but screw up by means of the internal square. Do not use any jointing material or washers on the joint between the tail pipe and the union nut, but make a metal-to-metal joint. Red-lead paint may be used on the threads.

Pipes and Valves.—Where the injector is fixed below its water supply, the valves required are the water cock and the main steam valve and feed-check valve. All pipes must be of full area, and elbows, sharp bends, and T-pieces should be avoided. A reduction of area or a restriction at one point is as harmful as if the whole pipe was too small. Pipes can be oversize without detriment. Excessive red-lead or other jointing material is liable to clog pipes and nozzles.

Steam Supply.—This should be taken from the highest point of the boiler to ensure dry steam. The pipe to the injector should be an independent one and supply no other apparatus. The pipe should be as short and straight as possible and superheated steam should not be used.

Water Supply.—It is essential that the suction pipe is absolutely airtight, as the slightest air leakage will affect the working of the injector. A fairly fine-mesh

strainer of ample area should be used. For high suction lifts a foot valve is advisable to facilitate starting. The inlet water should be as cold as possible, and the injector should be adjacent to the water supply.

Overflow.—The overflow pipe must be free and unrestricted. It should be of full area, short and straight, to allow for the free escape of steam at starting. This pipe plays an important part in the working of the injector, as, apart from the "sing" which gives the best evidence of a correct working, the only method of being certain that it is working "dry" is by watching the end of the overflow pipe. A good method is to allow for a break in the pipe by fitting a tundish or funnel, the pipe from which, if it is a size larger than the overflow pipe, may be as long as required. By this means the overflow can be seen and at the same time the escaping steam and water can be carried away freely at starting.

Before connecting up the injector the whole of the piping should be blown through to remove scale, red-lead, jointing, or dirt. Neglect of this precaution is a common cause of failure.

Starting and Stopping.—See that the feed-check valve is open. If the water supply is from overhead, open the water cock. See that the main steam valve is open. The injector is then started and regulated by means of the lever, which, when turned slightly to the left, admits steam and starts the injector working. Further movement to the left admits full steam, and it is then only necessary to find the position at which the injector works "dry" at the overflow without wasting. To shut off, shut the injector steam valve by turning the operating lever as far as it will go to the right. If injector is below water-level, shut water cock.

Pemberthy and White's Injectors

The Pemberthy Auto Injector is shown in Fig. 3. The details are quite clear: *R* is the steam jet; *S* the suction jet; *Y* the delivery jet, and *P* the overflow valve.

The Auto-positive, Fig. 4: *X* is the steam jet; *G* the suction jet; *H* the delivery jet; *L* the pressure valve, and *K* the vacuum valve.

White's Automatic Injector is shown in Fig. 5: *B* is the steam jet and *C* the combination cone. The above types of injector are manufactured by Whites-Nunan, Ltd.

Exhaust Steam Injectors

These are simply low-pressure injectors capable of working with steam at atmospheric pressure. An exhaust steam injector is capable of dealing with about 8 lb. of water per lb. of steam. Usually a supplementary steam jet at boiler pressure is provided. Cold feed water is essential if reliable and continuous operation is to be obtained.

Injectors for Vertical Boilers

Injectors for vertical boilers are frequently fitted as illustrated in Fig. 6. For pressure below 40 lb. per square inch they must be arranged non-lifting. This type of injector is shown in section in Fig. 7. By removing the bottom cap,

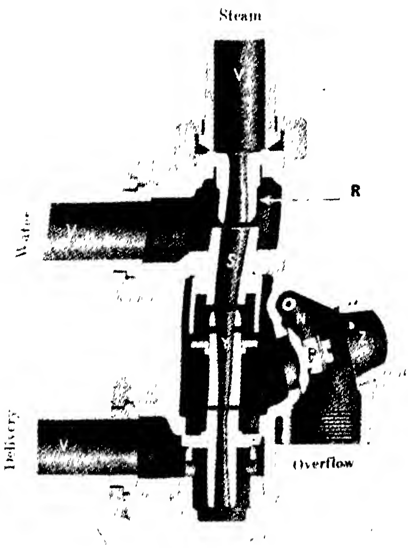


FIG. 3.—THE "PEMBERTHY" AUTO INJECTOR

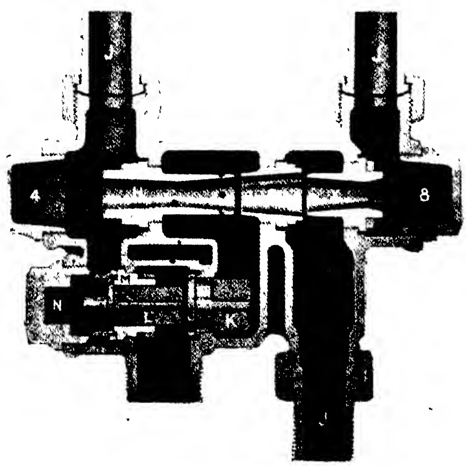


FIG. 4.—THE AUTO-POSITIVE INJECTOR

the delivery nozzle can be loosened with an ordinary spanner and taken out along with the jumper nozzle. This exposes the lifting tube, which needs a special tool for its removal. This injector is termed the "Premier" and is manufactured by Messrs. Holden & Brooke, Ltd. in five standard sizes for low, medium, and high pressures. The largest size is capable of dealing with 2,300 gallons per hour.

Ejectors and Water Lifters

The water ejector is a simplified injector and provides a reliable and economical method of raising small quantities of water from underground storage to overhead tanks. The "Pemberthy" ejector is shown in Fig. 8, in which *R* is the steam jet and Y the delivery jet. If the total lift is only small, the ejector can be fixed for lifting only as shown on the right-hand side of Fig. 9. Otherwise it should be fixed a few feet above the water level so as to force the water up the discharge pipe.

Injector Difficulties

If, when steam is turned on, the injector will not lift water, it may be due to:

(*a*) It is not getting its full supply of steam.

(*b*) Leakage of steam has made the water pipe hot, which prevents the injector from lifting.

(*c*) The inlet water is too hot.

(*d*) Steam nozzle or lifting tube blocked or choked with sediment. (See note (*c*) below.)

(*e*) Air leakage in water cock or water pipe.

(*f*) End of water pipe or strainer clogged or not covered by water in feed-water tank.

If injector gets water, but forces it through the overflow, the trouble may be:

(*a*) Defective steam supply.

(*b*) Inlet water too hot.

(*c*) Combining or delivery cones choked. (To remove sediment, wash out with a solution of one part muriatic acid and ten parts water.)

(*d*) Boiler-feed valve sticking.

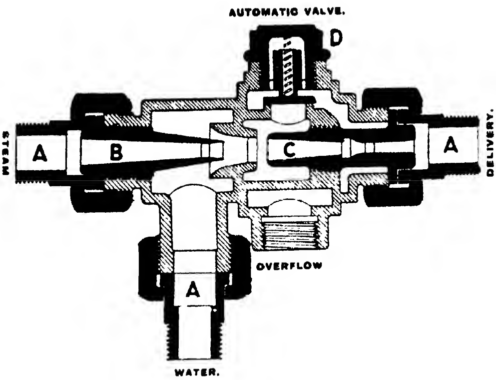


FIG. 5.—WHITE'S AUTOMATIC INJECTOR

CALORIFIERS

The term "calorifier" is applied to all types of tubular water-heating apparatus in which the heating medium, usually steam, but occasionally hot water, is admitted to a series of tubes and transmits heat to water surrounding the tubes. Two different types are manufactured, one termed the storage and the other non-storage, the main difference being in the size of the shell and holding capacity. Long experience has determined the most suitable design of this apparatus, and manufacturers have largely standardised the constructional details.

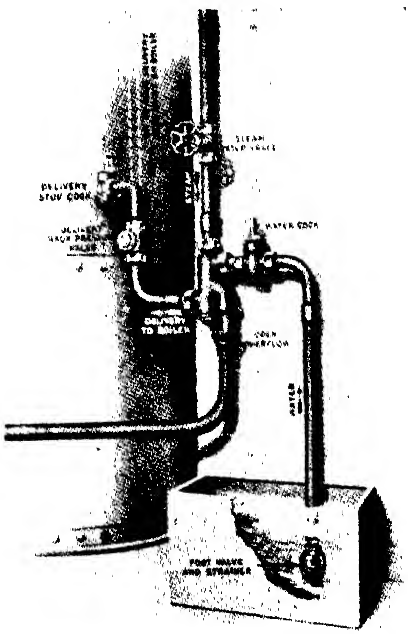


FIG. 6.—INJECTOR FEEDING A VERTICAL BOILER

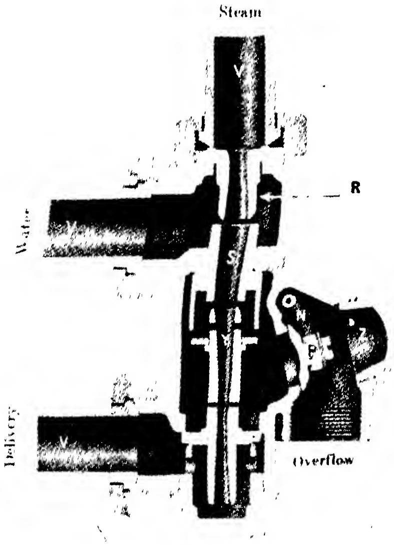


FIG. 3.—THE "PEMBERTHY" AUTO INJECTOR

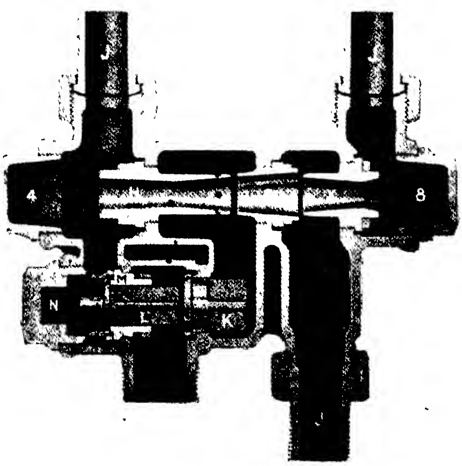


FIG. 4.—THE AUTO-POSITIVE INJECTOR

the delivery nozzle can be loosened with an ordinary spanner and taken out along with the jumper nozzle. This exposes the lifting tube, which needs a special tool for its removal. This injector is termed the "Premier" and is manufactured by Messrs. Holden & Brooke, Ltd. in five standard sizes for low, medium, and high pressures. The largest size is capable of dealing with 2,300 gallons per hour.

Ejectors and Water Lifters

The water ejector is a simplified injector and provides a reliable and economical method of raising small quantities of water from underground storage to overhead tanks. The "Pemberthy" ejector is shown in Fig. 8, in which *R* is the steam jet and Y the delivery jet. If the total lift is only small, the ejector can be fixed for lifting only as shown on the right-hand side of Fig. 9. Otherwise it should be fixed a few feet above the water level so as to force the water up the discharge pipe.

Injector Difficulties

If, when steam is turned on, the injector will not lift water, it may be due to:

(*a*) It is not getting its full supply of steam.

(*b*) Leakage of steam has made the water pipe hot, which prevents the injector from lifting.

(*c*) The inlet water is too hot.

body and steam chest are tested hydraulically to twice working pressure. Eighteen standard sizes are constructed; the smallest when supplied with steam at 50 lb. pressure will transmit 340,000 B.Th.U.s per hour and the largest 9¾ million B.Th.U.s when used as heating calorifiers. They can, of course, be used as hot-water calorifiers. This type is also manufactured with the body horizontal.

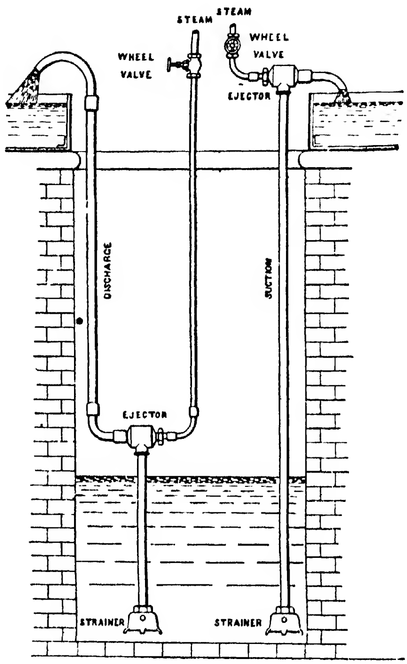


FIG. 9.—ARRANGEMENT FOR LIFTING AND FORCING

Heat Transmission

The rate of heat transmission in a calorifier is mainly dependent upon the temperature of the steam used for heating the water and the mean temperature of the water. It is usually assumed that 146 B.Th.U.s will be transmitted per square foot of heating or tube surface for each degree of mean temperature difference between the heating medium and the water. The formula used to find the heating surface in square feet is

$$\frac{L \times (t_2 - t_1)}{\left(T - \dfrac{t_2 + t_1}{2}\right) \times C} = \text{heating surface}$$

where t_1 = water inlet temperature.
t_2 = water outlet temperature.
T = temperature of heating medium.
C = a constant 146 for steam, 73 for hot water.
L = water to be heated in lb. per hour.

Example.—It is required to heat 1,000 gallons of water from 60° to 180° F. per hour by means of steam at 50 lb. gauge pressure. Taking 10 lb. as the weight of 1 gallon of water and 298° as the temperature of the steam, then

$$\frac{10{,}000 \times (180 - 60)}{\left(298 - \dfrac{180 + 60}{2}\right) \times 146} = 46 \text{ sq. ft.}$$

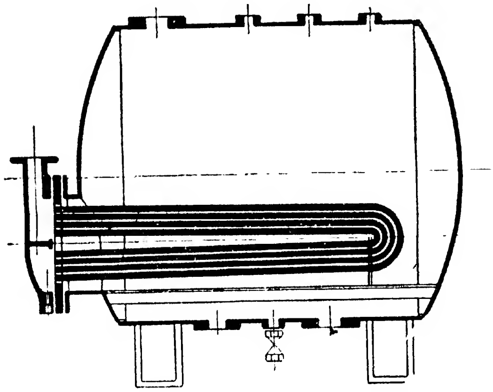


FIG. 10.—THE STORAGE TYPE OF CALORIFIER

WATER HEATERS

Hot water is required for many purposes, and is obtained very cheaply when low-pressure or waste steam can be obtained from engines, pumps, or other plant. Feed-water heaters are usually a necessity in most steam-raising plants, because large volumes of cold water entering a boiler tend to cool plates rapidly and set up unequal contraction and at the same time reduce the steam pressure, while hot water produces a better circulation and a more uniform temperature.

Two distinct types of water heaters are in use: (1) the closed or surface heater, in which the water to be heated is pumped through a series of tubes and the steam is admitted to a space surrounding the tubes; and (2) the direct-contact heater, in which steam is brought into direct contact with the water and condensed.

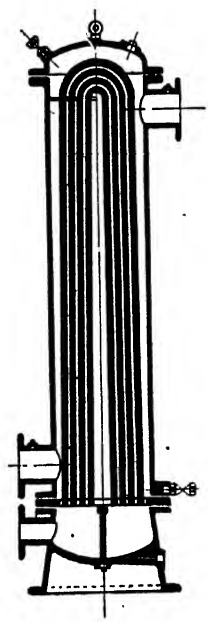


FIG. 11. — THE NON-STORAGE TYPE OF CALORIFIER

Steam for Water Heating

When steam enters a closed heater it gives up its latent heat at the entering pressure. The weight of steam to raise the temperature of a given quantity of water will be seen from the following example. Find the weight of steam at 5 lb. per square inch gauge pressure to raise the temperature of 1,000 gallons of water per hour from 60° to 180° F. The latent heat of steam at 5 lb. is 963 B.Th.U.s, and the heat required is $(180 - 60) \times 10{,}000 = 1{,}200{,}000$ B.Th.U.s approx. Then $1{,}200{,}000 \div 963 = 1{,}246$ lb. per hour.

Saving due to Feed Heating

The increase in boiler efficiency obtained by heating feed water with steam that otherwise would be wasted can be deduced from the formula

$$\frac{(T - t)\,100}{H - (t - 32)} = \text{increase per cent.}$$

where t = temperature of water entering heater.
T = temperature of water leaving heater.
H = total heat of steam at boiler pressure.

The original efficiency will then be increased by the percentage obtained.

Example.—Inlet temperature 60°; outlet temperature 180°; steam pressure 150 lb. (gauge); boiler efficiency 60 per cent. with feed at 60°. Let 1,201 = total heat of steam. Then

$$\frac{(180 - 60) \times 100 \times 75}{1201 - (60 - 32) \times 100} = 7{\cdot}6 \text{ per cent.}$$

By increasing the efficiency from 60 per cent. to 67·5 per cent., the saving in fuel consumption would be

$$\frac{67{\cdot}6 - 60}{67{\cdot}6} \times 100 = 10{\cdot}2 \text{ per cent.}$$

The Closed or Surface Heater

The closed or surface heater usually consists of a cast-iron cylindrical vessel containing a series of tubes of copper or brass expanded into tube plates or headers. The heater may be of vertical or horizontal design, and is installed with the water inlet connected to the discharge side of the boiler-feed pump.

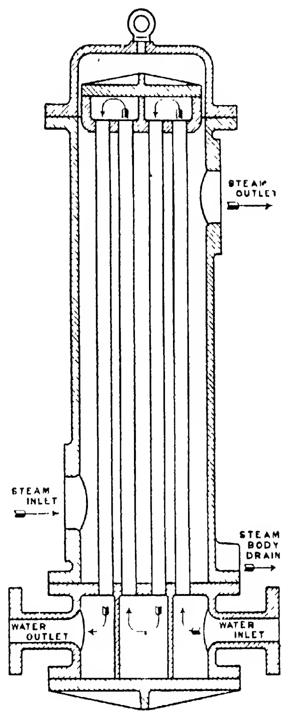


FIG. 12.—THE CLOSED OR SURFACE HEATER

A vertical type of heater is shown diagrammatically in Fig. 12. This represents a "Pearn Rapid" heater as manufactured by Messrs. Frank Pearn & Co., Manchester, and consists of a steam body of cast iron for saturated steam of low pressure and steel for high pressures and superheater steam. The tubes can be of copper, brass, or other material, and are expanded into a water box at one end and a floating head at the other. It will be seen that the water box, steam body, and floating head covers are removable without disconnecting either water or steam pipes. Twenty-four standard sizes are constructed, the smallest having a capacity of 200 gallons per hour and the largest 19,500 gallons. When these heaters are open to atmosphere and fed with low-pressure steam, the water can be raised to approximately 212° F. With greater steam pressures and a closed heater, higher water temperatures can be obtained.

The general arrangement of a closed heater using the exhaust steam from a horizontal steam-engine is shown in Fig. 13. It will be seen that the feed pump delivers cold water to the heater and forces it through the tubes and into the boiler. A vent pipe open to atmosphere prevents excessive back pressure.

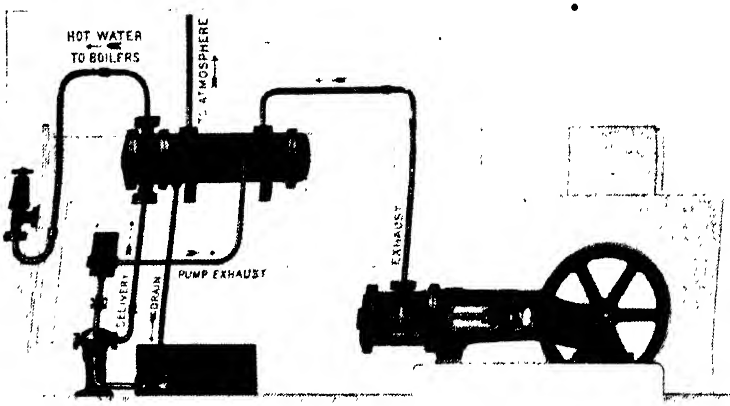


FIG. 13.—ARRANGEMENT OF A CLOSED OR SURFACE HEATER TAKING EXHAUST STEAM FROM A HORIZONTAL ENGINE

Direct-contact Heating

Direct-feed heating has the important advantage of liberating a large proportion of the air present in feed water and thus reducing the possibility of corrosion occurring in the boiler. Oil and grease, however, are quite as dangerous as air, and when the exhaust steam contains oil in any quantity it is unsuitable for feed heating, and an efficient oil separator should be used.

The weight of steam that must be used in a direct-contact heater in order to raise the temperature of a given quantity of water will be seen from the following example. It is necessary to increase the temperature of 100 gallons of water per hour from 60° to 200° F., using dry steam at 10 lb. per square inch by gauge. Taking 239 B.Th.U.s as the sensible heat of steam from 0° F. and 954 B.Th.U.s as the latent heat, then $\frac{1{,}000\,(200 - 60)}{954 + (239 - 200)} = 140$ lb. and the weight of the water and condensate will be 1,000 + 140 = 1,140 lb. If the dryness fraction of the steam is 0·9, then $10 \times 140 \div 0{\cdot}9 = 155$ lb. of steam will be required.

The Weir Direct-contact Feed-water Heater and De-aerator

A sectional view of the well-known "Weir" heater manufactured by Messrs. G. & J. Weir, of Glasgow, is shown in Fig. 14. The heating steam is taken from the low-pressure receiver of main engines, and the exhaust of auxiliary engines, such as feed pumps, electric light, and fan engines can be led into the heater through the non-return valve (*B*) on the side of the apparatus. A perforated cylindrical waist piece forming an annular steam space round the heater, and surmounted by a spray piece, also perforated, is fitted to ensure the uniform mixing of the steam and water. The boiler-feed pump delivers the feed water

into the heater through the spring-loaded valve (*D*) on the cover. The water is forced through this valve in a thin sheet, and is instantly heated by contact with the steam. The water is further broken up by passing through the spray piece, thereby bringing the greatest possible surface into contact with the steam.

As the pressure in the heater is generally much less than that of the entering water, the effect of this lowering of the pressure, and the sudden heating of the water, is to liberate the air in the water. This is removed to the condenser, or to the atmosphere, by way of the cock (*K*) on the air vessel placed on the top of the heater. The feed water is thus rendered to a large extent non-corrosive, and falls to the bottom of the heater at the boiling temperature corresponding to the pressure.

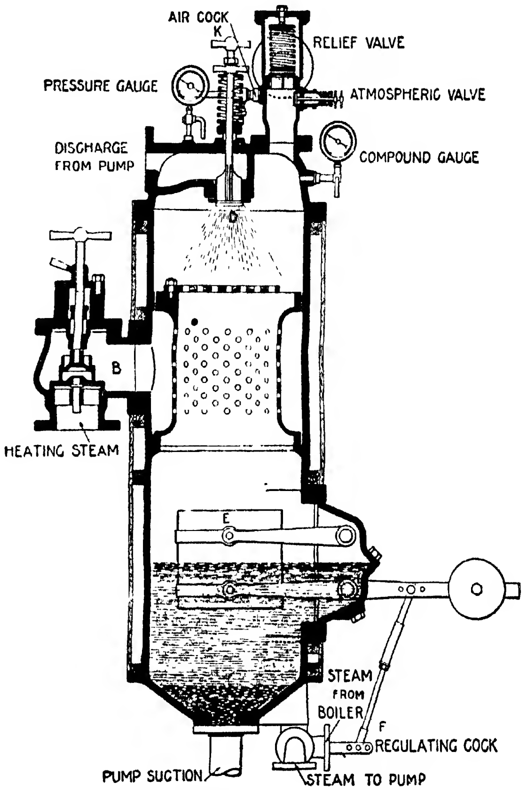


FIG. 14.—THE "WEIR" DIRECT-CONTACT HEATER AND DE-AERATOR

The combination of the automatic regulating gear with the heater has long been a special feature of the Weir apparatus. The float (*E*) shown in the lower part of the heater is a pan with water-tight bottom and sides, but open on the top. It is suspended on the two levers, so as to move up and down with a parallel motion; the bottom lever spindle is carried through the door at one end, and is balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance when one-half is immersed in water. The weight lever is connected by a rod to another lever, which actuates the regulating valve (*F*) and controls the supply of steam to the pump drawing from the heater. When the water in the heater rises the float is raised and the regulating valve opened,

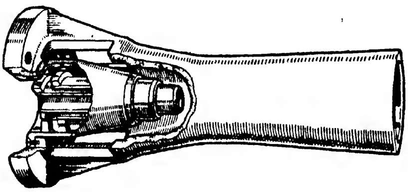


FIG. 15.—THE "TRICONE" HEATER

and when the water-level is lowered, the float follows and the valve is closed. The speed of the pump is thus regulated by the quantity of water passing through the heater.

A relief valve and two pressure gauges are fitted to the cover of the heater. The compound gauge shows the pressure in the heater, while the other gauge shows the pressure of the ingoing water; the latter pressure should be from 15 lb. to 25 lb. and should be regulated by the spring-loaded inlet valve.

Water Heating with Live Steam

Ejector heaters provide a ready and convenient method of heating liquids by steam up to boiling-point if desired and of keeping the heated liquid in

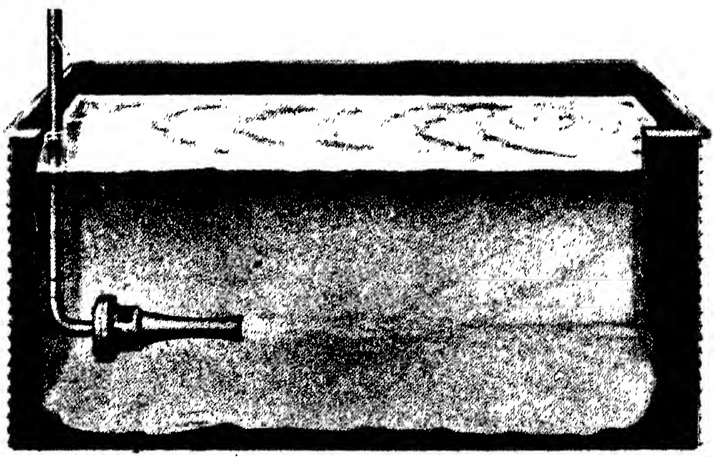


FIG. 16.—ARRANGEMENT OF HEATER IMMERSED IN A TANK

constant circulation. They are largely used in dyeing tanks, brewing vats, and other similar installations.

The characteristic feature of the "Tricone" heater shown in Fig. 15 and manufactured by Messrs. Holden & Brooke, Ltd., is silence and absence of the vibration usually experienced with the ordinary form of jet or nozzle heating apparatus.

The illustration shows the principle on which quiet operation is accomplished; with a single nozzle, noise will accompany the sudden disruptive action caused by a single cone of steam impinging on the water. In this heater steam is introduced in two thin layers and one fine jet. The nozzles are arranged concentrically; the outer and intermediate nozzles each pass a thin annulus of steam, while the centre one, being of relative small bore, passes a fine circular jet. Each nozzle projects farther from the centre than the one adjacent to it; thus the steam is condensed at three different points by the water sucked into the casing through the water ports which are located relative to the nozzles so as to take full advantage of the suction created by the jet.

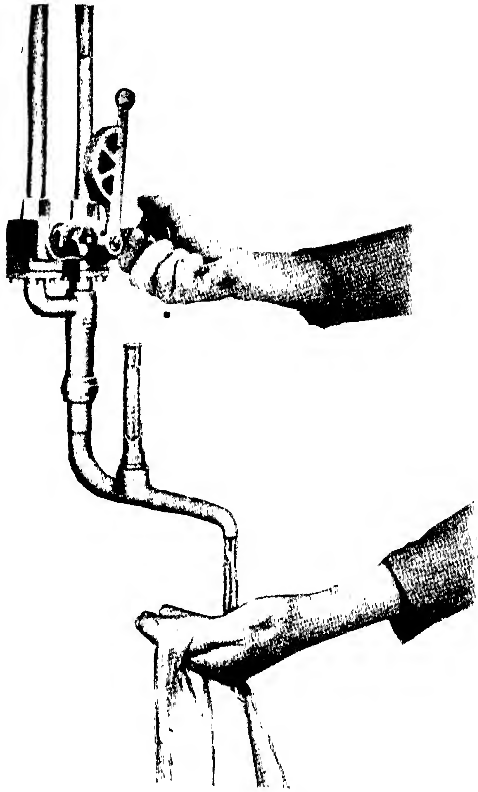


Fig. 17.—Apparatus for obtaining hot tap water

The heater is made in two styles: (*a*) immersion type for fixing in a tank or other vessel; and (*b*) a flanged type suitable for fixing in a pipeline. The former type is perfectly silent. The tank or container should be not less than three times the length of the heater for pressures up to 30 lb. per square inch, and not less than five times for higher pressures. The heater should be fixed as near to the bottom of the tank as possible, but to ensure satisfactory results should not be submerged to a greater depth in feet than the pounds per square inch steam pressure supplied to the heater. Successful installation depends upon reasonable rigidity both of the tank and of the fixing of the heater. The arrangement is shown in Fig. 16.

Hot Tap Water

Hot water for industrial washrooms and similar purposes can be very conveniently obtained by means of the "Baby" water heater shown in Fig. 17. This is a most compact apparatus and operates in silence without escape of steam. A safety guard prevents steam being turned on before the water and so prevents the possibility of scalding. The apparatus is supplied by the Cox Engineering Co., Ltd., and is reliable and easily fitted.

Another pattern of this heater, known as the "Junior Cox," is illustrated in Fig. 18. This is designed for use in the textile and beverage industries, dairies, distilleries, tanneries, chemical industries, etc., where hot water is used for process work and is instantaneously provided at the machine without storage.

It will be seen from Fig. 18 that the heat exchange chamber is fitted with porcelain balls or grids. This arrangement enables an instantaneous supply of water to be obtained without pulsations, spluttering, or escape of steam.

Steam required for Heating

The heating effect of the steam may, for practical purposes, be computed on the following basis: 1 lb. of steam will raise the temperature of 100 gallons of water by 1° F. To find the weight of steam, divide the gallons to be heated by 100 and multiply by the temperature rise.

Example.—Find the weight of steam to heat 2,500 gallons of water from 60° to 180° F. Then (2,500 ÷ 100) × (180 − 60) = 3,000 lb.

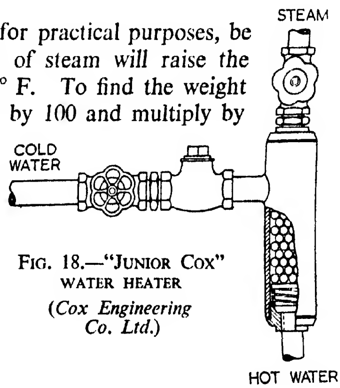


FIG. 18.—"JUNIOR COX" WATER HEATER

(*Cox Engineering Co. Ltd.*)

Water Mixtures

To find the temperature of 30 lb. of water at 60° F. mixed with 50 lb. of water at 160° F. Then

30 lb. of water contains 30 × (60 − 32) = 840 B.Th.U.s.
50 lb. of water contains 50 × (160 − 32) = 6,400 B.Th.U.s.
80 lb. of mixture contains 840 + 6,400 = 7,200 B.Th.U.s above 32° F.;
therefore 7,200 ÷ 80 = 90·5 + 32 = 122·5° F.

WATER DATA

One gallon of water at 62° F.	=	10·0 lb.
One gallon of water at 62° F.	=	0·16 cub. ft.
One gallon of water at 62° F.	=	277·27 cub. in.
One cubic foot of water at 62° F.	=	62·35 lb.
One cubic foot of water at 62° F.	=	6·23 gallons.
One ton of water at 62° F.	=	35·9 cub. ft.
One ton of water at 62° F.	=	224·0 gallons.

CAPACITY OF TANKS

Rectangular Tanks

Size.	*Gallons per Inch Depth.*	*Size.*	*Gallons per Inch Depth.*
3 ft. × 2 ft.	3·12	6 ft. × 3 ft.	9·36
3 ft. × 3 ft.	4·68	6 ft. × 4 ft.	12·48
4 ft. × 2 ft.	4·16	6 ft. × 5 ft.	15·60
4 ft. × 3 ft.	6·24	6 ft. × 6 ft.	18·72
4 ft. × 4 ft.	8·32	7 ft. × 4 ft.	14·56
5 ft. × 3 ft.	7·80	7 ft. × 6 ft.	21·84
5 ft. × 4 ft.	10·40	8 ft. × 4 ft.	16·64
5 ft. × 5 ft.	13·00	8 ft. × 6 ft.	24·96

Circular Tanks

Diameter.	*Gallons per Inch Depth.*
1 ft.	0·408
2 ft.	1·63
3 ft.	3·67
4 ft.	6·53
5 ft.	10·21
6 ft.	14·69
7 ft.	20·00
8 ft.	22·52

LIQUID FUELS AND LIQUID-FUEL FIRING

FEW engineers with practical experience of oil-fuel burning would burn solid fuels in preference to oil. The advantages of oil compared with solid fuels are many, and if oil fuels were as plentiful and could be obtained at a price equivalent to that of coal, little of the latter would be used for steam-raising purposes.

The advantages of liquid fuels are due to the increased efficiency owing to clean heating surfaces and superior combustion; convenience in handling, storing, and feeding furnaces; absence of ash, dust, and clinker; better control of firing conditions; reduction in staff, and absence of smoke and grit. The disadvantages are due in some degree to the danger of storing large quantities of oil fuel, the occasional choking of sprayers, and the extra cost in some cases compared with solid fuels.

Liquid Fuels

Natural mineral oils in the crude state consist of hydrocarbon compounds with impurities such as sulphur, water, and earthy matter. Crude oil can be used as a fuel, but its flash point is generally low and its viscosity high, and it requires to be used with great care. Usually the valuable petrol, illuminating, and lubricating oils are removed by distillation, and the residual liquid is termed "liquid fuel."

In addition to petroleum residue, whale oil, coal-tar oil, creosote, and pitch can all be satisfactorily used for steam-raising purposes in land installations.

COMPOSITION AND CALORIFIC VALUE OF LIQUID FUELS

Fuel	*Sp. Gr.*	*C*	*H*	*O*	*S*	*N*	*Gross Calorific Value B.Th.U.s*	*Flash Point °F.*
Borneo	0·96	86·7	10·6	2·5	0·03	0·05	18,800	225
Mexican	0·95	83·0	11·2	3·9	3·7	0·31	18,300	165
Rumanian	0·93	87·0	11·8	0·7	0·16	0·15	19,300	240
Admiralty Oil Fuel	0·928	86·4	11·5	1·7	0·34	—	17,900	185
Anglo-Persian	0·902	86·6	12·4	1·6	1·3	0·16	19,000	173
Scotch Shale	0·86	86·4	12·8	1·21	0·26	—	18,240	—

Flash Point

The flash point is the temperature at which the oil, if heated, slowly gives off sufficient inflammable vapour to ignite or flash with a blue flame on the application of a flame. The Pensky-Martens apparatus is largely used to determine the flash point, and consists of a jacketed vessel, the inner portion of which forms the oil chamber. A thermometer is suspended so that the bulb is covered with oil, and heat is obtained from a Bunsen burner in such a manner that the temperature rises 1° C. or 1·8° F. per minute. The flash point of liquid fuels varies from about 150° F. to 250° F.

Viscosity

Viscosity is a measure of fluidity or the resistance to relative motion. The coefficient of viscosity is usually termed simply the viscosity and is measured by means of a viscometer. The "Redwood" viscometer consists of an oil cup provided with an agate orifice or nozzle of standard dimensions. The cup is supported in an oil or water bath provided with a stirrer. The cup contains the oil to be tested to the height of a standard mark. The bath is heated by means of a Bunsen flame and a thermometer indicates the temperature of the oil. A graduated flask is placed underneath the oil cup, and the time in seconds for 50 c.c. to flow into the flask gives a direct reading in "Redwood" seconds. Viscosity should not exceed 2,000 seconds for a flow of 50 c.c. at 30° F.

The viscosity of oil fuels is important, because oils that do not flow freely may require to be heated in order to enable them to be transferred from storage to service tanks.

Fire Point

The fire point is the temperature at which the oil gives off sufficient inflammable vapour to continue burning. It is approximately 50° F. above flash point, but varies widely.

Settling Point

The temperature at which oil congeals and ceases to be fluid. This is usually below 30° F.

Specific Gravity

Specific gravity (sp. gr.) is the ratio of the mass volume of oil to the mass of an equal volume of water at 60° F. The specific gravity of oil fuels varies from about 0·8 to 1·0, and can be determined by means of a specific gravity bottle or with an hydrometer.

Calorific Value

The calorific value of a fuel is usually expressed in terms of British Thermal Units (B.Th.U.) and is the number of B.Th.U.s generated by the complete combustion of 1 lb. of fuel when the products of combustion are cooled to 60° F. A bomb calorimeter is invariably used for this test. Calorific values of

liquid fuels vary from about 18,500 to 19,500 B.Th.U.s per lb. and can be calculated approximately from the formula B.Th.U. per lb. gross = $18{,}650 + \frac{5{,}000\,(I - S)}{S}$, where S is the specific gravity.

Combustion of Oil

The theoretical quantity of air for complete combustion can be found approximately from $A = 0{\cdot}347\ H + 0{\cdot}115\ C$, where A = lb. of air, H = percentage of hydrogen, and C = percentage of carbon. *Example.*—$H = 12{\cdot}4$ per cent.; $C = 86$ per cent. Then: $0{\cdot}347 \times 12{\cdot}4 + 0{\cdot}115 \times 86 = 14{\cdot}1$ lb. of air.

Excess Air

As with all other fuels, an excess of air above the theoretical quantity is always necessary to ensure complete combustion. The maximum theoretical percentage of carbon dioxide (CO_2) in the products of combustion will depend upon the composition of the fuel, but for most practical purposes it can be said to average about 15·5 per cent. by volume. Thus, with 11 per cent. CO_2 in the waste gases the excess air would be $\frac{(15{\cdot}5 - 11)}{11} \times 100 = 41$ per cent.

Oil-firing Systems

Three systems are in general use for atomising and spraying liquid fuels into furnaces. These are the pressure jet, the steam jet, and the compressed-air jet system. The pressure system is probably the most economical method of spraying oil, but the steam system is lower in first cost, and where the oil consumption is comparatively small and the extra feed-water consumption of little consequence, the reduced first cost and the greater simplicity of the latter system may be a recommendation.

The pressure system comprises a horizontal or vertical fuel pump which draws oil from a service tank through a filter and discharges it, at a pressure dependent upon the construction of the burner and the size of the burner orifice, through an oil heater, where the oil is raised in temperature to approximately the flash point; then through a discharge filter to the burner, where it is atomised and sprayed into the furnace.

The steam-jet system is extremely simple, and consists of a burner using steam as the atomising agent, together with an overhead service tank containing a steam coil for heating the oil and facilitating its flow, a strainer, and the necessary connections. The oil supply to the burner is by gravity from a height of 15 ft. or more, and the steam pressure varies from about 5 lb. to 30 lb. per square inch. Approximately 4·5 per cent. of the total steam generated in a boiler is used by the burners in spraying oil. The question of extra steam and water consumption must therefore be taken into account.

The air-pressure system is largely used for industrial processes in smelting and refinery works. Air is supplied to the burners by means of a blower or air compressor at pressures of from 6 in. to 20 in. water gauge, but more frequently

a pressure of from 5 lb. to 30 lb. is used. With an air burner operating at 5 lb. per square inch the consumption of free air would be about 25 cub. ft. per minute while burning 150 lb. of oil per hour. When using burners of the air-pressure type, the oil must be of low viscosity when it reaches the burner.

Oil-fuel Installations

Oil-fuel installations using the pressure-jet system consist of a fuel pump, oil heater, filter, and the necessary burners, fittings, and piping, and are usually designed as a complete unit mounted upon a bedplate. Either two complete units are provided or a duplex plant is mounted upon a single bedplate.

The details of the burners vary widely, but are mostly simple in construction. Typical examples are illustrated.

Lighting-up Arrangements

In the case of small installations working on the pressure system, it is possible to start up from cold either by using a light oil that will atomise without special heating apparatus or by means of a supply of waste, wood, or a torch to heat the oil as it passes through a steel U-tube before reaching the burner. When lighting up is frequent, and with large installations, a hand or power pump is used to force the oil, under pressure, through an oil heater provided with a paraffin-oil burner, where it is heated before reaching the burners in the boiler furnace.

Working Conditions

In the pressure system the oil pressure may vary from about 50 lb. to 150 lb. per square inch according to the size of the diaphragm and orifice plate in the burner, the quality of oil it is desired to burn, and the number of burners in use. The quantity of oil is regulated by the fuel pump, and is controlled as required by the rate of combustion, usually by adjusting the relief valve on the pump, any excess being released back to the suction side of the pump. When more than one burner is in use, the control valves to the burners are normally kept wide open in order that an equal pressure shall operate all burners.

Oil Temperature

All except the lightest grades of oil require preheating. The temperature necessary to render an oil sufficiently fluid for spraying depends upon the character and viscosity of the oil. Heavy oils with a specific gravity of 0·9 or more may require a temperature as high as 200° F., but in no case should it exceed the flash point. Light oils such as shale will atomise at a temperature as low as 80° F. Excessive temperature may, and often does, cause pulsation.

Filters

Both suction and discharge filters should be periodically cleaned. These are either self-cleaning or in duplicate, so that one can be cleaned while the other is in use. Any material difference in the observed pressures shown by the gauges on the inlet and outlet sides of the filters will indicate the necessity for cleaning.

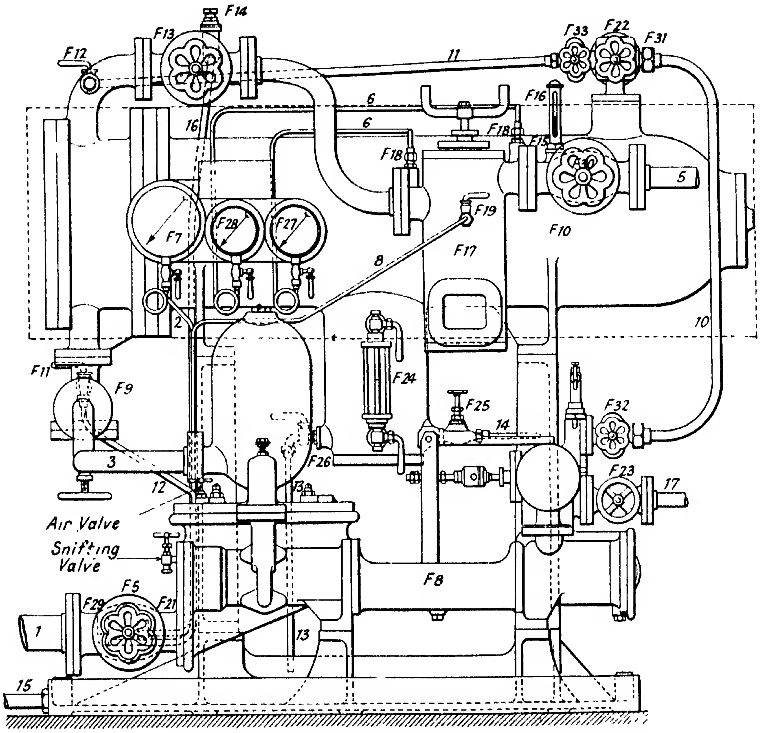


FIG. 1.—ARRANGEMENT OF WALLSEND-HOWDEN PUMPING AND HEATING UNIT

Air Distributors

Experience shows that the most successful way to admit air to a furnace when mechanical burners are used is to supply it all through an opening surrounding the atomiser. Poor combustion and smoke usually follow any attempt to supply air otherwise. Air distributors, directors, registers, or doors should be regulated to attain to the normal percentage of carbon dioxide, or until a light smoke is visible at the chimney outlet. Insufficient air will cause black smoke and the deposit of carbon in the furnaces.

Precautions

Heavy fuel oils are non-explosive, difficult to ignite, and incapable of spontaneous combustion. The vapour when mixed with air, however, is explosive, and it is always possible that a partly filled tank may contain an explosive mixture. Ignition may be caused by an open light, an electric or other spark, smoking, and an electrical short-circuit. Oil fires cannot be extinguished

by water or steam, and chemical extinguishers or sand should be used for this purpose.

The principal cause of accidents is the opening up of parts of an oil system or the removal of oil burners without first properly shutting down the oil supply.

GENERAL DESCRIPTION OF OIL-FUEL SYSTEMS AND BURNERS

The Wallsend-Howden Pressure System

In addition to storage and service tanks of heating, storing, and providing oil for ready use, this installation consists of a pumping and heating unit shown in Fig. 1. The oil is drawn from a service tank through a strainer by means of a Weir oil-fuel pump *F*8 and discharged to heater *F*10, where its temperature is raised to reduce the viscosity sufficient to enable a fine spray to be obtained at the oil-fuel burner. After leaving the heater the oil is forced through a discharge strainer *F*17, in which the oil is finely strained.

The fittings and their accessories are generally mounted on an oiltight tray to form the complete unit. These units have the advantage of being complete and compact, but with all parts readily accessible, and being erected thus at the maker's works, the amount of work to be done on site and therefore the costs of installation are reduced to a minimum.

Two units, each capable of dealing with the oil required at full power, are usually installed, so that one provides a stand-by in case the working unit requires cleaning or overhaul. The various fittings and pipes are given in the following lists:

LIST OF FITTINGS

No.	*Description*
F 5	Valve, Pump Suction
F 6	Tail Coupling for Press and Vacuum Gauge Pipe
F 7	Compound Pressure and Vacuum Gauge
F 8	Weir's Simplex Oil-fuel Pump
F 9	Valve, Heater Inlet
F 10	Oil-fuel Heater
F 11	Cock, Drain from Heater (Steaming Out)
F 12	Cock, Steam to Heater (Steaming Out)
F 13	Valve, Heater Outlet
F 14	Valve, Heater Escape
F 15	Thermometer Pot
F 16	Thermometer
F 17	Auto-Klean Discharge Strainer
F 18	Tail Couplings for Pressure-gauge Pipes
F 19	Cock, Discharge Strainer Drain
F 21	Non-return Valve, Discharge Strainer Drain
F 22	Valve Box, Steam to Installation
F 23	Valve, Exhaust from Pump

LIST OF FITTINGS (*continued*)

No.	*Description*
F 24	Gauge Glass and Guard on Water Collector
F 25	Valve, Water Collector Drain to Drain Tank
F 26	Cock, Water Collector Drain to Tray
F 27	Strainer Inlet, Pressure Gauge and Cock
F 28	Strainer Outlet, Pressure Gauge and Cock
F 29	Non-return Valve, Heater Escape to Suction
F 30	Installation Master Valve
F 31	Stud Coupling Steam to Pump
F 32	Control Valve, Steam to Pump
F 33	Control Valve for Steaming Out Heater

LIST OF PIPES

No.	*Description*
1	Oil Fuel Pump Suction
2	Suction to Compound Gauge
3	Oil-fuel Pump Discharge to Heater
4	Heater Outlet to Discharge Strainer
5	Discharge to Burners
6	Discharge Strainer to Pressure Gauge
8	Discharge Strainer Drains to Suction
9	Steam to Installation
10	Steam to Pump
11	Steam to Steaming Out Heater
12	Drain from Steaming Out Heater
13	Drain from Water Collector to Tray
14	Drain from Water Collector to Drain Tank
15	Drain from Tray to Sump
16	Heater Escape to Suction
17	Exhaust from Pump

The Heater

The construction of the heater is shown in Fig. 2. Oil passes through solid-drawn steel tubes, bent in U-form and expanded into one steel-tube plate, thus diminishing the risk of leaky tubes. As the tubes are free to expand under the influence of temperature variations, a connection for blowing steam through the tubes has been introduced to clean them without the necessity of taking any part adrift, and without fear of slackening the tubes in the tube plate. To deal with such a contingency as solid matter adhering to any part of the tubes, a supply of special cleaning tools is available for scraping the straight portion of the tubes and also for passing round the bends.

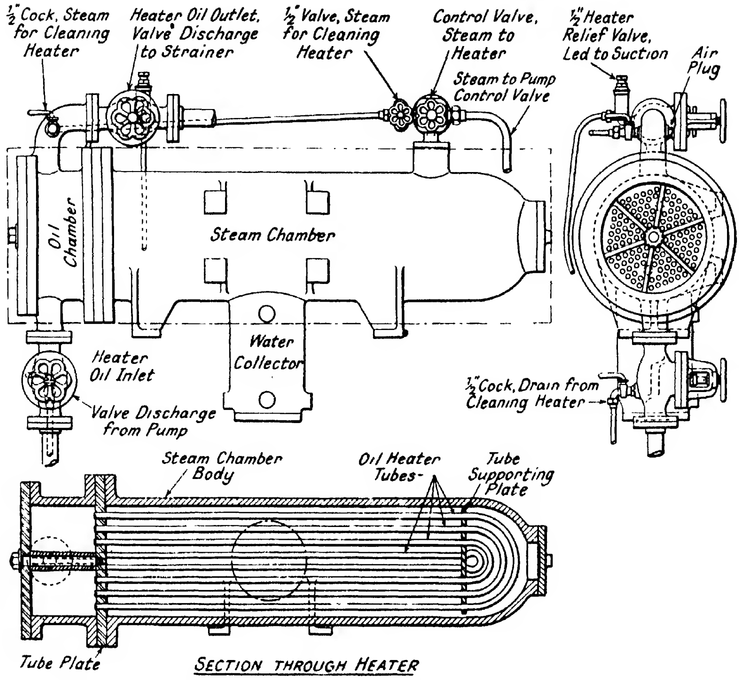


FIG. 2.—THE WALLSEND-HOWDEN STANDARD U-TUBE OIL-FUEL HEATER

Strainers

Strainers are a most important component of any oil-fuel firing system, and are frequently fitted in pairs with change-over valves, so that one acts as a stand-by while the other is in use. From the point of view of safety and convenience this system has the objection that if the change-over valves are not perfectly tight on their seats, owing to dirt or for any other reason, hot oil under pressure and in a highly inflammable condition may escape when a joint in the strainer is broken for cleaning purposes. In order to eliminate this possible source of danger, the Auto-Klean strainer has been introduced as a standard part of the equipment. The straining medium of the cleaner is cleaned by turning a handle and no joints have to be broken.

The strainer is shown in part section in Fig. 3. Oil enters by the inlet branch and then passes between a series of perforated steel plates accurately spaced apart, but with only a fine clearance between each plate, and assembled in the form of a cartridge, which can be revolved by means of a handle.

Inserted between the plates, but independent of them, is a series of fixed knives or cleaners. Any dirt entering the strainer with the oil lodges on the

outside of the cartridge; if, therefore, the accumulation of dirt becomes excessive, there would be a drop in pressure indicated by the pressure gauges as the oil passes through the strainer. In order to clean the cartridge when there are indications that it is dirty, the handle is given a half-turn so that the knives which remain stationary remove the dirt from the cartridge and cause it to fall into the sump.

The Oil-fuel Burners

After leaving the strainer the oil travels through the discharge pipes to the burners. The burner is shown in section in Fig. 4 and consists of four parts, viz. the body, cap, nozzle, and diaphragm. Typical nozzle ends are shown enlarged and in detail in Figs. 5 and 6, which illustrate two different types—the first is known as the "F" type and the second the Intermediate type. In both the oil passes nearly tangentially through the diaphragm into the swirling chamber formed in the nozzle. The velocity with which it passes through the nearly tangential holes causes the oil to spin or revolve in the swirling chamber, so that on issuing from the nozzle the oil flies apart under the influence of centrifugal force and opens out into the form of a conical spray of mist-like particles.

The wall of the hollow cone of spray should be as thin and as uniform as possible so that air can mix rapidly with it. It will be understood from the foregoing that the centrifugal action which atomises the oil depends very largely upon the velocity with which all the oil passes through the holes in the diaphragm, and that, therefore, the fineness and angle of spray may be changed by varying the size or number of holes in the diaphragm. The smaller the total area through the holes in the diaphragm relative to the area of the hole through the nozzle, and the higher the velocity with which the oil passes through the holes in the diaphragm, the wider and finer the spray will be.

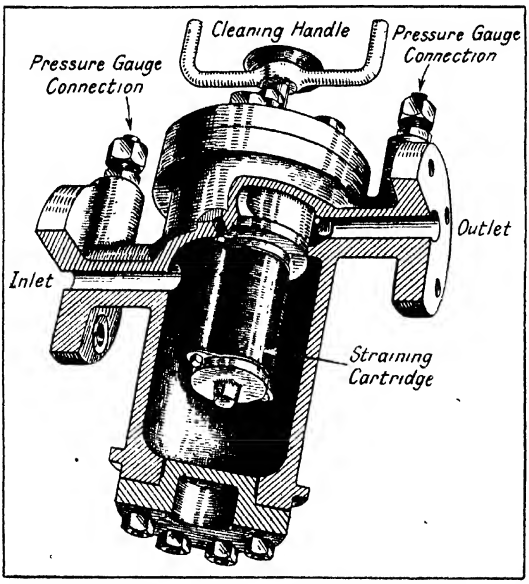


FIG. 3.—SECTION THROUGH WALLSEND-HOWDEN OIL-FUEL STRAINER

It is obvious that

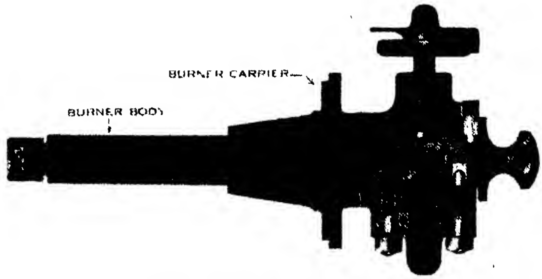


FIG. 4 (*right*).—SECTION THROUGH WALLSEND-HOWDEN OIL-FUEL BURNER

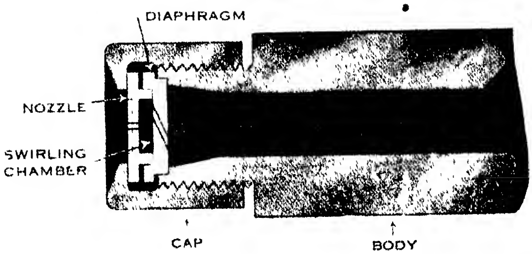


FIG. 5 (*left*). — ENLARGED VIEW OF WALLSEND-HOWDEN BURNER END — "F" TYPE

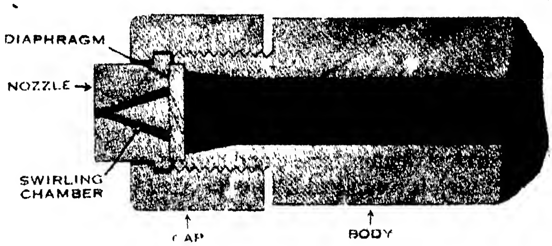


FIG. 6 (*right*).—ENLARGED VIEW OF WALLSEND-HOWDEN BURNER END—"INTERMEDIATE" TYPE

the velocity of the oil increases with pressure. The "F" type of nozzle and diaphragm works at pressures varying between 75 and 175 lb. and gives the widest spray. It is suitable for all grades of oil, including the heaviest and most viscous.

The "Intermediate" type of burner works at pressures between 50 and 100 lb. It is suitable for central firing with fire-bars removed with most classes of oil fuels, and also for furnaces in which oil is being burned with the fire-bars remaining in position, provided the diameter of the furnace is suitable. Tables are available showing approximate quantities of oil passed through the burners per hour when provided with nozzles and diaphragms of a given size.

The Air Director

The main difference in the problems of burning oil and coal lies in the fact that a large portion of the latter is solid at the temperature of ignition. Oil, however, is sprayed into the furnace in the form of a mist and partial vapour, which the draught catches up and carries with it, so that unless the air is admitted to the furnace in such a manner that will cause rapid and intimate mixture of air and oil, some of the oil will escape unburned.

The natural-draught air director is illustrated in Fig. 7, and consists essentially of two concentric trunks with vanes fitted in the annular space between them through which the main air supply is admitted to the furnace. The vanes impart a whirling motion to the air, so that in boilers with circular furnaces the air travels along a spiral path which is also followed by the fine

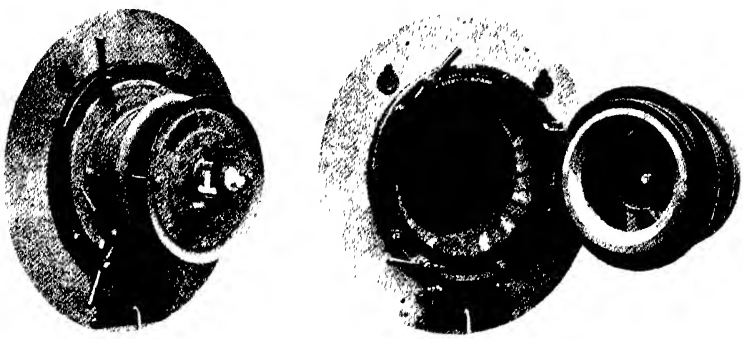


FIG. 7.—NATURAL-DRAUGHT WALLSEND-HOWDEN AIR DIRECTOR HINGED TO GIVE EASY ACCESS TO FURNACE

particles of oil carried by the air. This has the effect of retaining or banking up the burning vapours, gases, and carbon near the front of the furnace, so that combustion is completed before the gases leave the intensely hot zone.

The sliding sleeve can be adjusted to control the amount of air entering the furnace. The function of the burners in the front plate is to regulate the position of the flame by admitting air to the inner trunk. Closing the louvres brings the flame forward, opening them throws it back. The amount of opening is dependent upon the quantity of oil being burned and upon its quality and calorific value.

In addition to imparting a whirling motion to the air, the vanes in the director perform other important functions. Owing to radiation from the furnace, the vanes become heated and serve the double purpose of preventing loss due to radiation and of heating the air which comes into contact with them, thereby assisting combustion.

Oil Temperature

Most fuel oils require to be heated in order to reduce their viscosity, or, in other words, increase the fluidity, sufficient to ensure fine atomisation. The

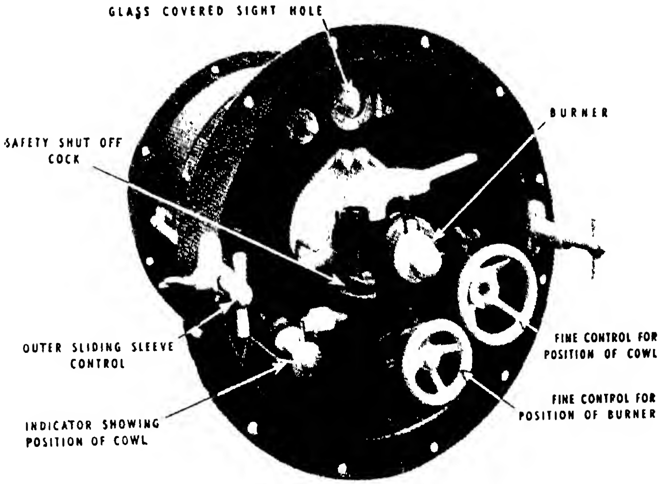


FIG. 8.—FINE CONTROL AIR DISTRIBUTOR FOR OPERATING UNDER FORCED DRAUGHT

This type of air distributor is installed on the largest type of water-tube boilers, and is suitable for a capacity up to 2,500 lb. of oil per hour. (*The Wallsend Slipway & Engineering Co., Ltd.*)

thinner the oil, the finer will be the spray produced. The greater the viscosity, the more difficult it becomes for the burner to break up the oil into particles fine enough for its intimate mixture with the air necessary for combustion.

With the Wallsend-Howden system, thin light oils of specific gravity 0·92 and under may be efficiently atomised at temperatures of about 150° F. over a large range of pressures from low to high.

Medium oils of specific gravity above 0·92 and below 0·95 can be efficiently atomised at temperatures about 200° F. or less, and over a large range of pressures from medium to high.

The heaviest and most viscous oils require more heat, but the maximum temperature is limited, and therefore to obtain fine atomisation recourse must be made to working at high pressures.

By low pressures is meant pressures of 50 lb. per square inch or less. Moderate pressures cover the range from about 50 lb. to 100 lb. Any pressure above this is considered high, but pressures up to 200 lb. can be safely used, as all discharge fittings in the Wallsend-Howden system are designed to withstand a test pressure of 400 lb. per square inch.

The temperature at which various fuel oils can be efficiently burned is shown in Fig. 9, and it will be noted that the viscosity should be reduced by heating to the required temperature, thus ensuring efficient burning so far as the class of fuel is concerned. The classes of fuel shown are:

The Air Director

The main difference in the problems of burning oil and coal lies in the fact that a large portion of the latter is solid at the temperature of ignition. Oil, however, is sprayed into the furnace in the form of a mist and partial vapour, which the draught catches up and carries with it, so that unless the air is admitted to the furnace in such a manner that will cause rapid and intimate mixture of air and oil, some of the oil will escape unburned.

The natural-draught air director is illustrated in Fig. 7, and consists essentially of two concentric trunks with vanes fitted in the annular space between them through which the main air supply is admitted to the furnace. The vanes impart a whirling motion to the air, so that in boilers with circular furnaces the air travels along a spiral path which is also followed by the fine

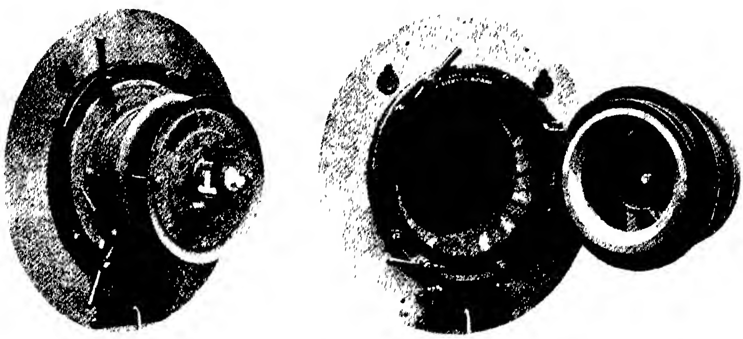


FIG. 7.—NATURAL-DRAUGHT WALLSEND-HOWDEN AIR DIRECTOR HINGED TO GIVE EASY ACCESS TO FURNACE

particles of oil carried by the air. This has the effect of retaining or banking up the burning vapours, gases, and carbon near the front of the furnace, so that combustion is completed before the gases leave the intensely hot zone.

The sliding sleeve can be adjusted to control the amount of air entering the furnace. The function of the burners in the front plate is to regulate the position of the flame by admitting air to the inner trunk. Closing the louvres brings the flame forward, opening them throws it back. The amount of opening is dependent upon the quantity of oil being burned and upon its quality and calorific value.

In addition to imparting a whirling motion to the air, the vanes in the director perform other important functions. Owing to radiation from the furnace, the vanes become heated and serve the double purpose of preventing loss due to radiation and of heating the air which comes into contact with them, thereby assisting combustion.

Oil Temperature

Most fuel oils require to be heated in order to reduce their viscosity, or, in other words, increase the fluidity, sufficient to ensure fine atomisation. The

superintendent engineer or chief engineer can obtain about three spots on the curve which can be plotted on the chart and the best working temperature and pressure estimated.

Changing Burners

In the event of any burner requiring to be extinguished and removed for cleaning or other purposes, the air regulators should at once be closed to prevent air impinging on the hotplates.

Spare burners should be kept ready made up for use with nozzles, and diaphragms fitted so that only about half a minute is occupied in changing a burner.

The spray should be fine and clear, and should spread at an angle of not less than 70°.

If the spray is streaky, or the angle is less than 70°, the burner is not working properly and must be changed and cleaned.

The burner valves are not for regulating the quantity of oil delivered by the burners, and must be either fully open or shut. The output is regulated by oil pressure.

Cleaning Nozzles and Diaphragms

When it is necessary to clear the holes of the nozzles and diaphragms, a soft copper wire should be used for the purpose, as harder material may damage the orifices.

Paraffin should be used for washing the nozzles, diaphragms, and burners, as it readily removes asphalt, etc., and care must be taken to see they are perfectly clean.

Carbon forming in the Furnace

Coke may form in the furnace, owing to too low an oil temperature, a dirty burner not spraying properly, or the oil spray being too wide for the furnace. Whatever the reason, the coke should be broken up and pushed to the back of the furnace, where it will burn away. Excessive coke formation is a nuisance, and should be avoided, but slight coke formation does not lower the efficiency provided it is not allowed to accumulate. Its formation varies with different classes of oil, so that the angle of spray must be adjusted as found necessary. The wider the spray that can be used without excessive coke formation, the higher the efficiency of combustion.

A satisfactory method of dealing with excessive coke formation in a furnace is to deliver a small quantity of fresh water into the furnaces along with the oil. This can be done by introducing the water at the snifting valve on the pump.

Pulsation

Pulsation or panting may be caused by any or all of the following conditions: lack of air, excessive oil temperature, poor arrangement of flues, and water in

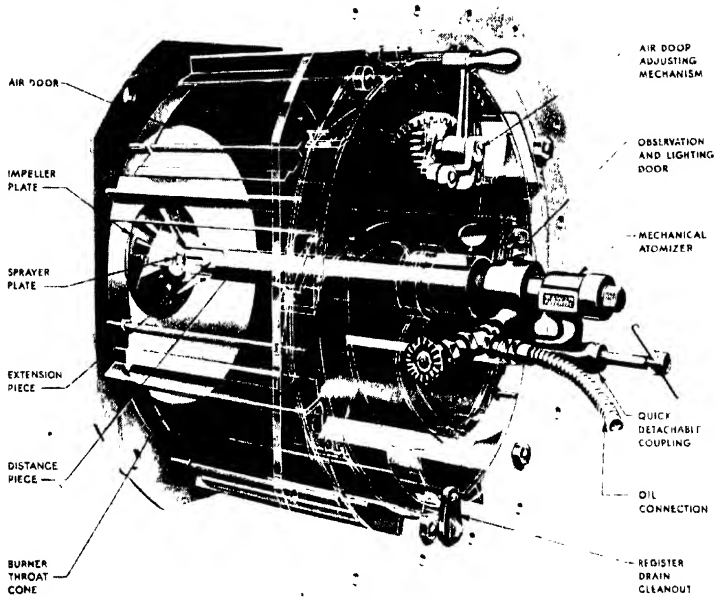


FIG. 10.—SINGLE-FRONT AIR REGISTER ADAPTED FOR FORCED-DRAUGHT CONDITIONS
(*Babcock & Wilcox, Ltd.*)

the oil. This vibration can usually be remedied by careful attention to oil temperature and pressure and air supply.

Babcock & Wilcox Air Registers and Burners

The air register (Fig. 10) is of circular construction and adapted for forced-draught conditions. The air control consists of a mild-steel framework mounting a number of inward-opening mild-steel air doors on the circumference. The front is completely enclosed by a mild-steel screen plate which minimises the effect of direct radiation from the furnace, and on this plate are mounted the oil-fuel sprayer supports and the actuating gear for the air doors.

There are two vital requirements for the successful burning of fuel oil so far as the oil burner itself is concerned. They are:

(1) The oil must be completely atomised to a very fine mist.

(2) The atomised oil must be thoroughly mixed with sufficient air for complete combustion, and the maximum turbulence of the mixture obtained.

In Babcock & Wilcox burners the first condition—complete atomisation—is secured by forcing oil at the proper degree of fluidity through slots in the

inner face of a sprayer plate and discharging it tangentially into a conical chamber having an orifice at its apex. This causes the oil to whirl at high speeds around the conical chamber and to emerge from the orifice into the furnace in a hollow cone of minute particles in proper condition for mixing with the heated combustion air.

Experience shows that the second requirement is best met by the admission to the furnace of all the air necessary for combustion through a burner register and throat surrounding the atomiser. Air is admitted to the furnace through adjustable doors on the register, causing the air to rotate as it moves towards the atomiser tip. An impeller plate or diffuser at the furnace end of the atomiser

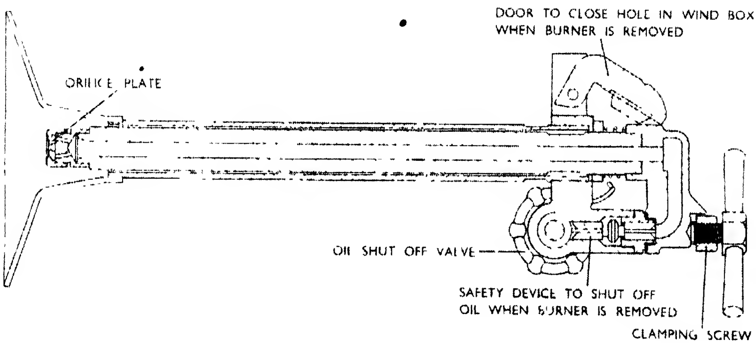


FIG. 11.—BABCOCK & WILCOX PATENT OIL BURNER

permits the passage of the proper quantity of air for mixing with the oil as it emerges from the sprayer plate. This results in a very short flame at all outputs.

Babcock & Wilcox Burners

Babcock & Wilcox burners are all fitted with a quick-release clip comprising a bridle and clamping screw, so that the burners can be easily and quickly dismantled and withdrawn. They are also fitted with safety automatic valves, which prevent oil leakage in the case of a burner valve being opened with the burner removed. The atomiser is in two parts: the distributing plate and the orifice plate. The distributing plate incorporates a number of channels varying from two to four, which are cut tangentially to a central race. One end of this race is closed by the back of the distributing plate, the other by the orifice plate.

The arrangement of the channels or slots produces a rotary action in the oil inside the race, the effect of which is that after discharging from the orifice, the oil is discharged in a conical spray of fine formation.

These burners are suitable for the heaviest grade of fuel oil, providing it is preheated to a temperature at which viscosity is suitable for spraying. This is usually 80–120 seconds Redwood No. 1.

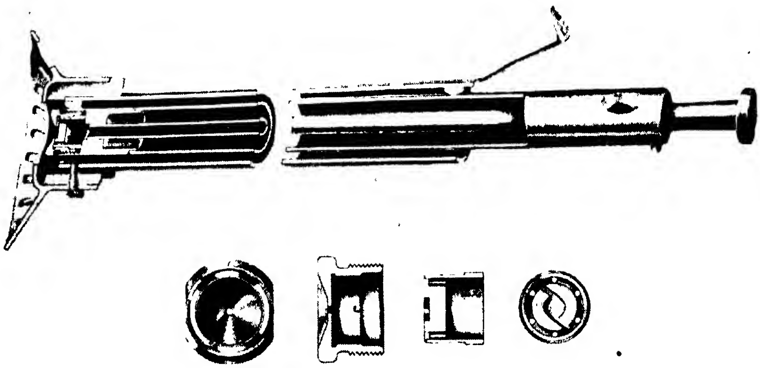


FIG. 12.—THORNYCROFT FUEL BURNER
Showing details of lightweight sprayer body, sprayer cap, and atomiser

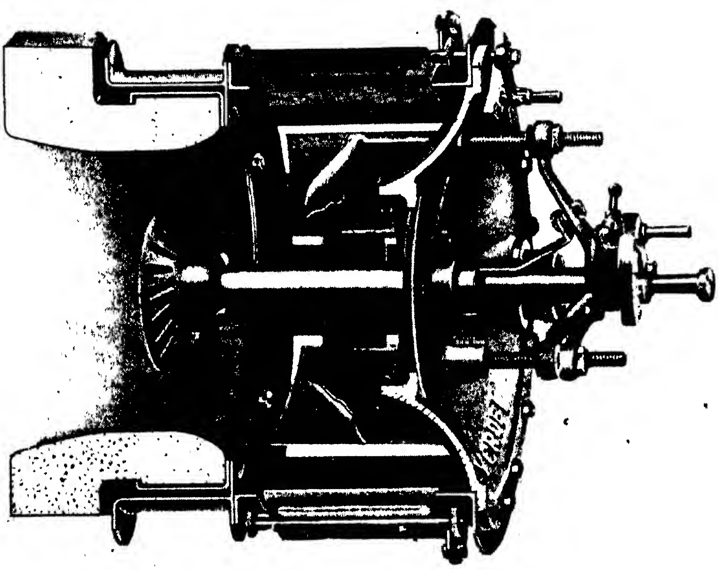


FIG. 13.—THORNYCROFT FUEL BURNER
Illustrating arrangement of furnace front and the throat.

The Thornycroft, Burner

The burner shown in Figs. 12 and 13 is suitable for use with a wide range of liquid fuels, including the heaviest boiler fuel oil and also creosote pitch.

The fuel is pumped under pressure into the central supply tube of the sprayer body via a flexible hose, safety cock, and quick-change clamp. It then passes into the atomiser whirling chamber through channels arranged tangentially to the outer diameter and finally out through the exit hole in the atomiser cap.

By these means the stream of fuel is caused to rotate at a very high velocity just before leaving the exit hole, after which it opens out into a wide conical cloud of extremely fine particles.

The combustion air, which enters the circular box surrounding the burner, is caused to rotate and develop into a vortex flow, attaining its maximum intensity at the point where it meets the fuel particles.

The output of individual burners is adjustable within limits by means of pressure changes, but each burner is supplied with quickly interchangeable atomisers of different capacities.

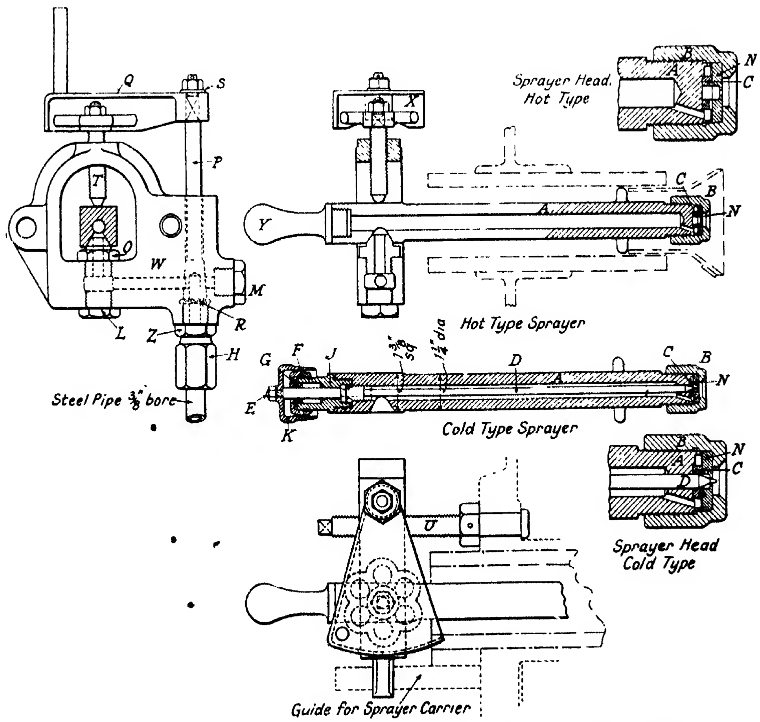


FIG. 14.—"HOT" AND "HOT OR COLD" OIL-FUEL BURNERS (*J. Samuel White*)

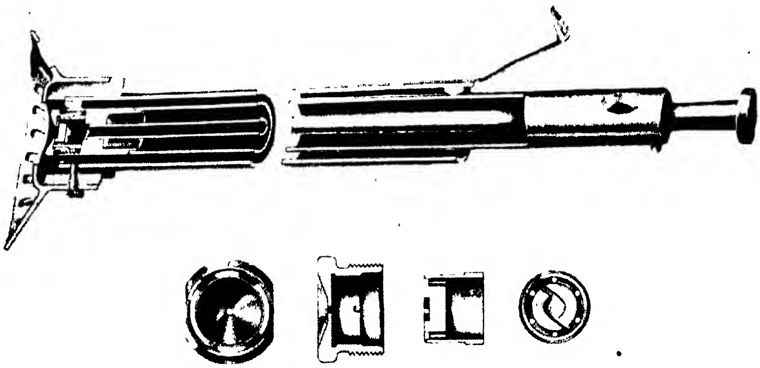


FIG. 12.—THORNYCROFT FUEL BURNER
Showing details of lightweight sprayer body, sprayer cap, and atomiser

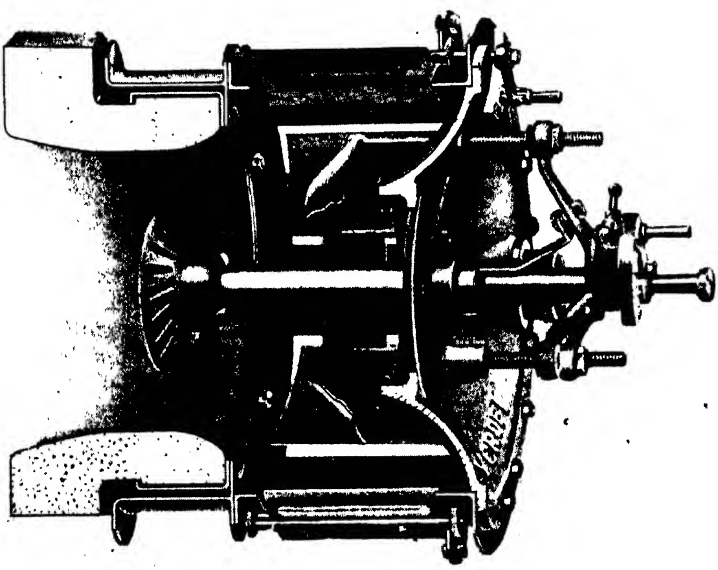


FIG. 13.—THORNYCROFT FUEL BURNER
Illustrating arrangement of furnace front and the throat.

prevents fluctuations of pressure in the burner supply line, and the pressure in this line is regulated by the adjustable spring-loaded relief valves, on the discharge side of the pumps, which can be set to any required pressure to suit the steaming conditions. The suction and discharge filters are each mounted on separate oiltight trays, which enables each installation to be laid out in the most convenient arrangement to suit the circumstances.

Apart from the general compactness of the unit, the chief difference in the general design is in the oil-fuel heater, which is of the large-bore coil design, the oil being on the inside of the pipe coil and the steam on the outside. Thus, the oil passes through the heater at a constant speed. The great advantage of this arrangement is that very little carbonisation of the oil takes place in the coils, provided the usual care and attention is given to the unit. Another important point is that, as the coils are made of solid-drawn steel tubing with no internal joints, there is little possibility of oil leaking through the steam side.

The furnace fronts manufactured by this firm follow general principles, both for natural-draught and forced-draught installations, care being taken to ensure that the air for combustion is easily regulated and is admitted in equal quantities all round the burner, to ensure complete mixture with the oil spray and perfect combustion.

The "White" oil burner is designed to atomise oil of all gravities up to the heaviest crude oil at pressures between 60 lb. and 95 lb. per

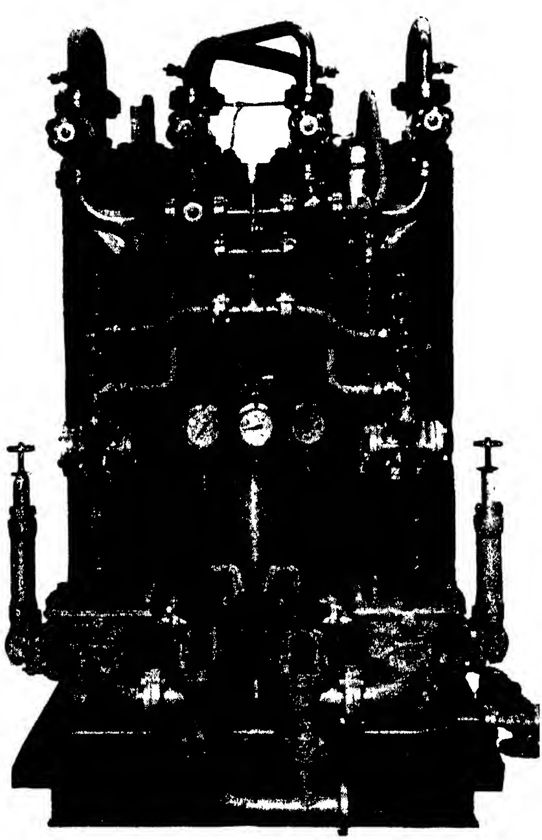


FIG. 15. — DUPLEX OIL PUMPING AND HEATING UNIT (*White's Marine Engineering Co., Ltd.*)

square inch. The oil, after being heated to obtain the desired viscosity for effective atomisation, passes along grooves in a cone surface, whence it enters the whirling chamber, where a pressure velocity conversion takes place, causing the oil to issue from the orifice in the form of a finely divided conical spray.

The latest-type "F" burner is shown in Figs. 16 and 17. It will be seen that the burner cannot be removed until the safety cock handle is in the "shut" position, the guard on the handle protecting the pinching screwhead, and, having removed the burner, the safety cock cannot be opened until it is replaced.

Todd Oil-burning Equipment

The Todd type "R" rotary burner shown in Fig. 18 is of the horizontal rotary atomising type, in which atomisation is accomplished by feeding the oil

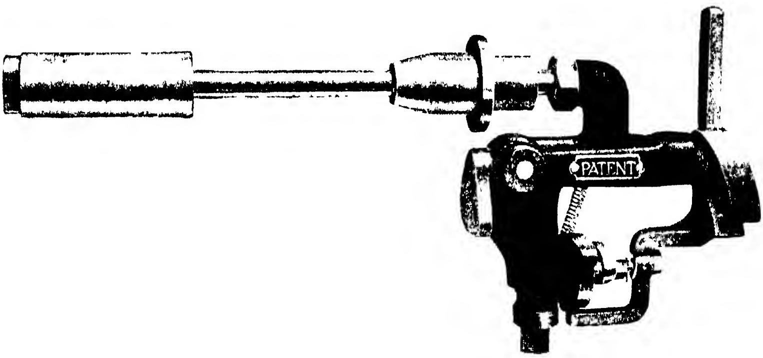


FIG. 16.—TYPE "F" OIL BURNER WITH SAFETY SHUT-OFF COCK
(*White's Marine Engineering Co., Ltd.*)

into an open horizontal cup, rotating at high speed by belt drive from an electric motor. Primary air for atomisation is supplied by a fan mounted on the same shaft as the spinning cup and driven by the same motor. Secondary air for combustion is admitted below the burner through a hinged air door.

The essential parts are the atomiser cup, air cone, the primary air fan, the motor, and the secondary air admission door.

The burner is constructed with a hollow hinge pin to enable it to be swung open when not in operation. Each burner is supplied with a special brick venturi ring, with an angle of recession designed to prevent impingement of the flame.

The motor drives the fan and spinning cup shaft through V-belts, at a speed of 3,400 r.p.m., and a simple arrangement on the motor seating is provided for adjusting the tension of the V-belts.

Air is admitted to the primary air fan through a small register on the underside of the burner, and is regulated by a butterfly damper installed in the

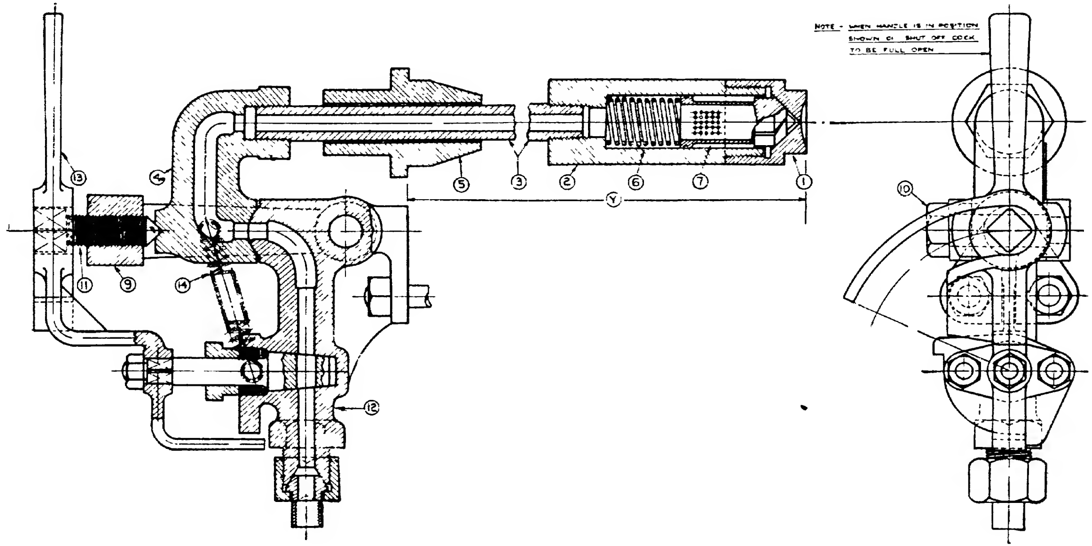


FIG. 17.—SECTIONAL VIEW OF TYPE "F" OIL BURNER WITH SAFETY SHUT-OFF COCK

1. Burner tip
2. Burner body.
3. Burner tube.
4. Burner head.
5. Burner guide.
6. Burner spring.
7. Strainer atomiser.
9. Union yoke.
10. Shackle bolt.
11. Pinching screw.
12. Quick detachable union with safety cock, complete with plug and glands.
13. Safety-cock handle.
14. Spring.

(White's Marine Engineering Co., Ltd.)

register; secondary air for combustion is admitted through the secondary air door, which is bolted to the cast-iron front plate.

The type "R" burner will operate with oil supplied by gravity, or, if more convenient, by a separate motor-driven rotary oil pump fitted with relief valves.

The burner is equipped with a vaporstat, operated by air pressure from the fan. This automatically shuts off the oil supply to the burner in the event of motor failure, or stoppage of the burner for any other reason.

Steam or Air Atomising Burners

The Todd steam or air atomising oil burners are designed in two sizes, small and large type, to burn any grade of fuel. The smaller type, shown in cross-section in Fig. 19, is designed to handle quantities of fuel up to 100 lb. per hour, and the larger type for any requirements above that amount, each with corresponding steam or air pressure, varying from 5 lb. to 20 lb. per square inch.

The oil is atomised by either steam or air, depending on the conditions and the grade of fuel to be burned. Air for combustion is obtained by natural draught, and simple control is provided to adjust the air supply to obtain maximum economy of operation.

The burners operate efficiently on the heaviest grades of fuel oil, preheated to a temperature of approximately 150° F., and steam is recommended as the atomising agent with this grade of oil. The burner will operate with equal efficiency on diesel oil or light grades of fuel oil, using either steam or air as the atomising agent.

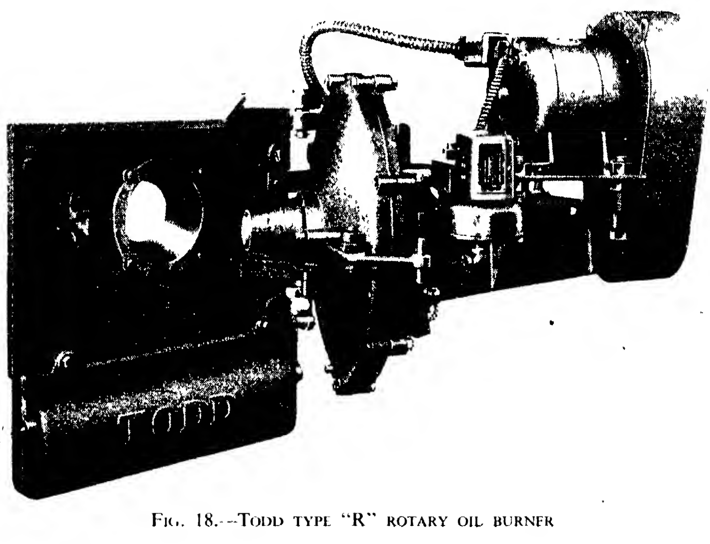


FIG. 18.—TODD TYPE "R" ROTARY OIL BURNER

The oil supply to the burners may be arranged from a gravity tank, situated at a suitable height, to ensure a steady flow of oil, or alternatively with a small pump, according to the space available and the output required. If the oil is too heavy a quality to be burned cold, then a suitable preheating arrangement must be provided, either by means of a steam coil in the supply tank, or alternatively by an oil heater inserted into the oil-supply pipe to the burner.

The Todd small-type steam or air atomising oil burner is particularly suitable for oil firing cooking ranges, bakers' ovens, roasters, metal-melting and pipe-bending furnaces, and small steam boilers, where the amount of fuel required to be burnt does not exceed 100 lb. per hour.

Mechanical Pressure Burner

The Todd "Vee Cee" mechanical-pressure atomising fuel-oil burner (Fig. 20) is designed for use in installations where it is desired to atomise solely by means of fuel-oil pressure and when the load fluctuates over a wide range.

The burner operates without the use of an atomising medium, such as steam or compressed air, and there are no moving parts. This type of burner, used in conjunction with Todd air registers, is particularly applicable to boilers of 100 h.p. and over, operating at 50 lb. working steam pressure or higher, such as water tube or horizontal return tubular boilers in both stationary and marine installations, petroleum refining shell and tube stills, and cracking coils. These burners may be installed with any of the Todd air registers.

The operation of the "Vee Cee" burner is simple and positive. Oil at

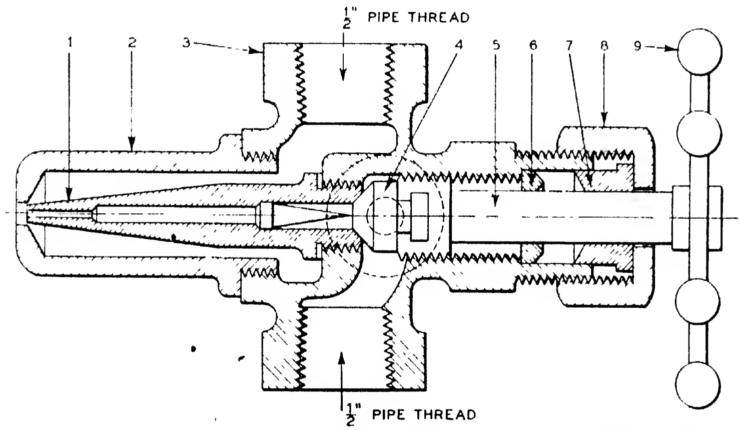


FIG. 19.—CROSS SECTION OF TODD SMALL-TYPE STEAM OR AIR ATOMISING BURNER

1. Oil nozzle.
2. Nozzle casing.
3. Body.
4. Swivel valve.
5. Valve spindle.
6. Neck ring.
7. Gland.
8. Gland nut.
9. Hand wheel.

register; secondary air for combustion is admitted through the secondary air door, which is bolted to the cast-iron front plate.

The type "R" burner will operate with oil supplied by gravity, or, if more convenient, by a separate motor-driven rotary oil pump fitted with relief valves.

The burner is equipped with a vaporstat, operated by air pressure from the fan. This automatically shuts off the oil supply to the burner in the event of motor failure, or stoppage of the burner for any other reason.

Steam or Air Atomising Burners

The Todd steam or air atomising oil burners are designed in two sizes, small and large type, to burn any grade of fuel. The smaller type, shown in cross-section in Fig. 19, is designed to handle quantities of fuel up to 100 lb. per hour, and the larger type for any requirements above that amount, each with corresponding steam or air pressure, varying from 5 lb. to 20 lb. per square inch.

The oil is atomised by either steam or air, depending on the conditions and the grade of fuel to be burned. Air for combustion is obtained by natural draught, and simple control is provided to adjust the air supply to obtain maximum economy of operation.

The burners operate efficiently on the heaviest grades of fuel oil, preheated to a temperature of approximately 150° F., and steam is recommended as the atomising agent with this grade of oil. The burner will operate with equal efficiency on diesel oil or light grades of fuel oil, using either steam or air as the atomising agent.

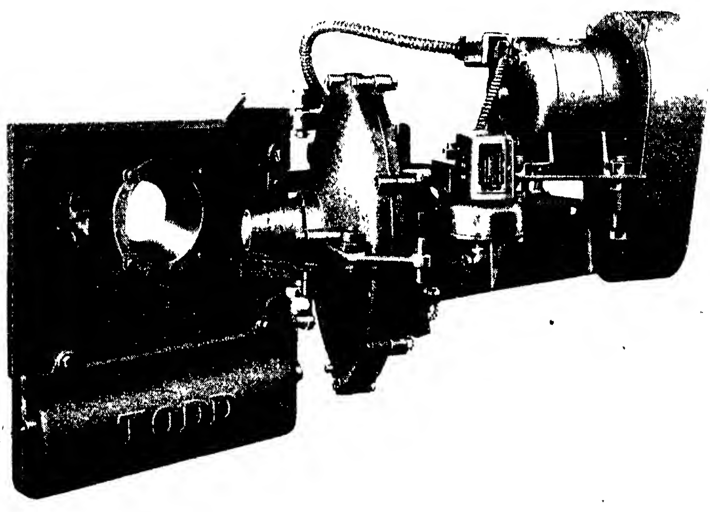


FIG. 18.—TODD TYPE "R" ROTARY OIL BURNER

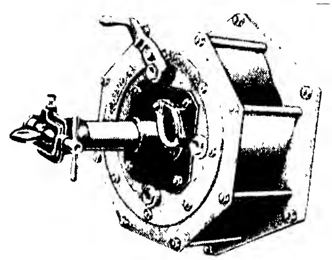


FIG. 21 - TODD "HEX-PRESS" BURNER

The atomiser handle conducts oil from the connection in the housing to the atomiser barrel; the fluted projection on this casting assists in handling the atomiser. The oil passes from the duct in the handle through the atomiser barrel to the nozzle body. A collar on the nozzle body prevents air from blowing around the spray.

The sprayer plate consists of a central chamber to which the oil passes by way of nearly tangential slots. Passage of the oil under pressure through these grooves causes the oil to rotate in this chamber, from which it passes to the orifice, leaving the latter in the form of a spray.

The sprayer plates are made in two types. One type has a flat leaving face and the other is made with a countersunk leaving face. The sprayer plates with the flat leaving face are generally for use on cylindrical boilers, and the sprayer plates with the countersunk leaving face are usually adopted for use on water-tube boilers.

The rear surface of the sprayer plate, which is machined and polished, makes a metal to metal joint with the outer face of the nozzle body. The sprayer plate is held against the latter by the sprayer-plate nut.

The Register.—The Todd "Hex-Press" register is of the all-round air type. Air flow and intermixture are produced by curvilinear surfaces embodied as integral parts of the air admission doors. These doors admit air around the oil spray practically entirely in front of the diffuser, the latter being at the fireroom end of the air register.

The scoop-like vanes are set at an angle which causes the deflected air streams to intersect the fuel issuing from the atomiser in a vertical plane adjacent to the flame cone or diffuser that is carried by the burner jacket tube.

This object is accomplished by forming the register wall of a series of overlapping louvres or hinged doors, which are opened inwardly to admit streams of air over their outer surfaces, and curved vanes carried respectively by the hinged doors, these

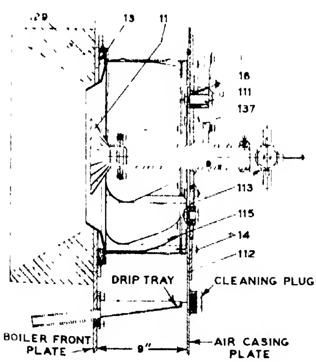


FIG. 22.- CROSS SECTION OF TODD "HEX-PRESS" BURNER

The numbered parts refer to the air register unit.

11. Diffuser.	112. Front plate.
13. Base frame.	113. Centre plate.
14. Column studs.	115. Air shutters.
16. Ring roller, plain	129. Venturi brick quarl.
111. Operating lever pin.	
137. Ignition door cover.	

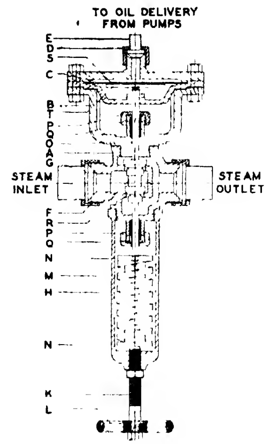


FIG. 23.—SECTIONAL VIEW OF TODD PUMP GOVERNOR

A. Chest.	*L*. Adjusting wheel.
B. Dome.	*M*. Spring.
C. Oil chamber.	*N*. Spring caps.
D. Oil connection nut.	*O*. Spindles.
E. Oil connection tail.	*P*. Gland nuts.
F. Steam connection nut.	*Q*. Gland rings.
G. Steam connection tail.	*R*. Valve.
H. Spring dome.	*S*. Diaphragm.
K. Adjusting screw.	*T*. Piston.

vanes being provided with curvilinear or scoop-like inner surfaces that direct the air streams into a rearwardly directed spiral air flow.

Pump Governor

The duty of a pump is to displace a certain volume of fluid within a given time to a certain point or at a given pressure.

The Todd pump governor (Fig. 23), when attached to the steam supply of any pump, will not only cause that pump to stop and cease work the moment any predetermined pressure has been obtained, but will start the pump again as soon as any predetermined drop in pressure has taken place, and not till then. This means that if a certain number of burners are shut down, the governor automatically comes into operation and reduces the supply of steam to the pressure pump. On the other hand, if an extra number of burners are brought into operation, the governor will again automatically increase the steam supply to the pressure pump. The oil pressure to the burners is thus regulated by the pump governor valve, thereby obviating the constant regulating of the steam valve on the pump to suit varying conditions.

E. P.

THE TESTING OF LUBRICATING AND FUEL OILS

OIL is one of the most important commodities of our present age, so much so that the life of the modern world would be impossible without it. It may be a matter for philosophical conjecture whether it would have been better for mankind had there been no inventions like the internal-combustion engine, gas turbine, and all the multitudinous mechanisms which surround us to-day, but the fact remains that our present-day industries, transport, and communications depend for their existence on supplies of oil.

Oils are available with a wide variety of properties, and they must be selected for the specific purposes for which they are intended if they are to fulfil their functions in the best possible way. A lubricating oil which is suitable for a certain set of conditions may be absolutely worthless as a lubricant under other conditions. For example, the type of oil which is required to operate at high temperatures in a steam-engine cylinder must have entirely different properties from an oil which lubricates the bearings of a light mechanism at normal atmospheric temperatures.

The various properties of oils, such as density, viscosity, flash-point, etc., can be ascertained by means of specially designed apparatus, and the combined results of these tests will enable an oil technician to determine for what purpose an oil is suitable. Various oils may be blended together so as to get combinations of certain properties, and the chemist to-day is able to produce oils of an enormous range and combination of properties.

Sources of Oils

Oils may be derived from three sources:

(1) Animal.

(2) Vegetable.

(3) Mineral.

Examples of animal oils are lard, sperm oil, and tallow. The fat is extracted from the body of the animal under pressure by steam. Vegetable oils include olive, castor, palm, colza, linseed, etc. These oils are removed from the seeds by either a pressing process or by means of solvents. Many of the animal and vegetable oils are unsuitable for lubricating purposes, as they are subject to gumming or rapid oxidation. These are often used for painting oils, where the drying and thickening properties are of value. Small quantities of animal or vegetable oils are, however, often blended with mineral oils.

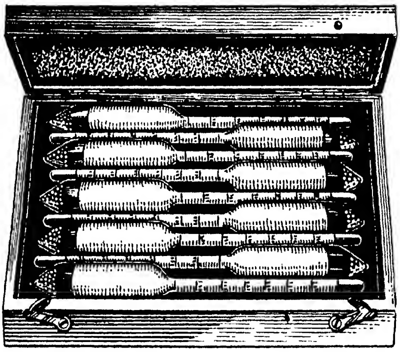


FIG. 1.—SET OF HYDROMETERS

The hydrometers in the set shown cover a wide range of densities.

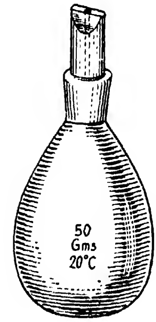


FIG. 2.—SPECIFIC GRAVITY BOTTLE

Mineral oils form by far the largest source of our supply. The tremendous extent of the petroleum industry and the enormous wealth involved in obtaining oil from the earth are of such importance as to exert an influence on international political issues.

How an Oil Lubricates

It is sometimes asked why a lubricant is necessary. This question will be dealt with briefly before going into the matter of the actual testing of oils.

If two metal surfaces are rubbed together, friction is generated between them. If the surfaces are very rough, this friction is high, due to the projections of the surfaces rubbing against each other. Friction generates heat, and if the pressure between the surfaces is sufficiently great and the temperature generated by the friction is high enough, seizure of the two surfaces may take place.

The purpose of a lubricating oil is to separate surfaces which are in relative movement by the formation of an oil film between them. So long as an oil film is maintained between the two surfaces they cannot seize. It is only when the oil film is broken and the surfaces come together that seizure can take place. The pressure forcing the surfaces together tends to squeeze out the oil film, and an oil must be chosen which will successfully resist this tendency under the conditions of operation. The oil must be "thick" enough to prevent the film being broken, but yet at the same time it must not be so "thick" as to cause excessive fluid friction.

In most cases the oil film is maintained by the relative motion of the surfaces, and when this motion ceases the oil is squeezed out. The "thickness" of the oil, or its viscosity as it is termed, changes with the change of temperature, the oil becoming "thinner" as the temperature rises. The viscosity at the working

temperature must therefore be such as to ensure that the oil film is maintained at that temperature. Therefore, oils working under high temperatures are very "thick" at normal temperatures, whereas oils working at low temperatures, as in refrigerators, are very thin at normal temperatures.

Testing of Oils

Specifications are laid down which define what the properties of various oils should be to meet the requirements of definite operating conditions. When a bulk supply of oil is purchased, it is therefore possible to get a representative sample and subject the oil to the approved test to see if it complies with the specifications. An obvious method of testing an oil is to subject it to the exact conditions under which it will work in practice and carefully note its behaviour over a long period. It is often impracticable, however, to carry out full-scale tests of this description, since failure of the lubricant may result in extensive and costly damage being done to the machine being lubricated. Similarly, a short test of this description may be very misleading, as it may not be possible to reproduce those factors which depend on the length of time of operation. The usual tests carried out are as follows:

Specific Gravity

The specific gravity of a substance is the ratio of the weight of a given volume of that substance to the weight of the same volume of water. The specific gravity of oils may be determined by two main methods—one by using a hydrometer, and the other by using a specific gravity bottle. The use of a hydrometer is the simpler method, and while this is not as accurate as the specific gravity bottle, it is sufficiently accurate for most purposes. A hydrometer consists of a glass float with a graduated stem weighted at the lower end so as to float vertically in a fluid.

When the hydrometer is placed in a liquid, it sinks to a depth which is dependent on the density of the liquid, and the specific gravity is read on the graduated stem at the point which coincides with the surface of the liquid. Hydrometers are often supplied in sets, each hydrometer being graduated to deal with a small range of densities and the total range of the set being dependent on the number of hydrometers in the set. Fig. 1 shows a case containing nine hydrometers which cover a range of specific gravities from 0·650 to 1·100 with a range of 0·05 each.

Fig. 2 illustrates a specific gravity bottle. This is a bottle which is fitted with a ground stopper through which is a small hole. The method of finding specific gravity is as follows: the bottle is first carefully cleaned and dried and then weighed with the stopper. It is then filled with distilled water, the stopper being inserted so as to ensure that the bottle is completely full, the small hole allowing water to pass through it as the stopper is inserted. After carefully wiping, the bottle is again weighed. The difference between the two weighings gives the weight of water. The bottle is then emptied of water, dried, and filled with

the oil to be tested, the stopper being inserted as before to ensure that the bottle contains the standard quantity. The full bottle of oil is then weighed, the actual weight of oil being found by deducting the weight of the empty bottle. The specific gravity is then calculated from $\frac{\text{weight of oil}}{\text{weight of water}}$.

In repetitive tests, the weight of the bottle empty and full of water will have been obtained (or this will be indicated by the maker on the bottle), so that all that is necessary for subsequent tests is to fill the bottle with the oil to be tested to enable the above calculation to be made. The standard temperature at which specific gravity should be taken is 60° F.

If a liquid is heated, it expands and therefore becomes less dense, which shows itself in a reduction of specific gravity. If it is not convenient to test oil at 60° F., the specific gravity may be corrected by adding 0·00034 for each degree F. below 60° F., or by subtracting 0·00034 for each degree F. above 60° F. For example, if the specific gravity of a certain oil at 100° F. is 0·880, then at 60° F. its specific gravity will be:

$$\begin{aligned} & 0{\cdot}8800 + (100 - 60) \times 0{\cdot}00034 \\ & = 0{\cdot}880 + 40 \times 0{\cdot}00034 \\ & = 0{\cdot}880 + 0{\cdot}0136 \\ & = 0{\cdot}8936 \end{aligned}$$

If two oils with different specific gravities are mixed, the specific gravity of the mixture can be calculated from the formula

$$Sm = \frac{V_1S_1 + V_2S_2}{V_1 + V_2}$$

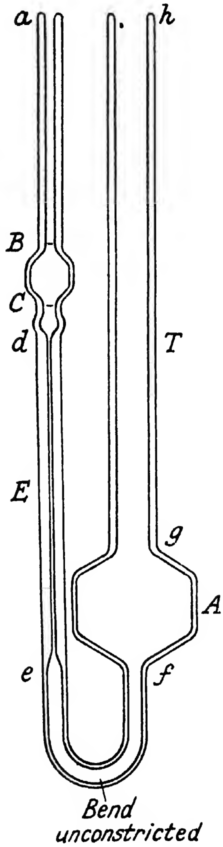
where:—

Sm is the specific gravity of the mixture.
V_1 is the volume of No. 1 oil.
V_2 is the volume of No. 2 oil.
S_1 is the specific gravity of No. 1 oil.
S_2 is the specific gravity of No. 2 oil.

The specific gravity of lubricating oils varies from about 0·86 and of fuel oils from about 0·65 to just over 1·0. The specific gravity is relatively unimportant from the point of view of the quality of lubricating oils; its main use is in the conversion of weights and volumes when oil is being purchased. In the case of fuel oils, the specific gravity gives some indication of the calorific value.

Viscosity

As already indicated, viscosity is one of the most important properties of a lubricating oil. The instrument for determining viscosity is called the viscometer. Fig. 3 illustrates a U-tube viscometer. British Standard Specification No. 188—1937 specifies the use of such an instrument. The U-tube viscometer is immersed in a bath of liquid at a controlled temperature. A given volume of oil is placed in the bulb A down the tube T, and this is sucked up the small capillary tube E.



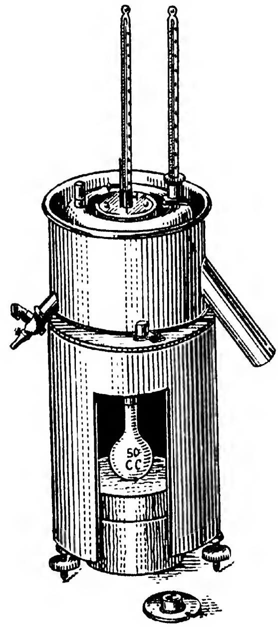


FIG. 3 (*left*).—A U-TUBE VISCOMETER

FIG. 4 (*above*).—THE REDWOOD VISCOMETER

The oil level is adjusted to the point *B*, and the oil is then allowed to flow back through the capillary tube under its own head. The time for the oil level to fall from *B* to *C* is taken, and from this the viscosity is calculated.

Another type of viscometer is used in which a sphere of specified diameter and density is allowed to fall through the liquid contained in a flask and the time is taken for the sphere to fall a specified distance. Calculations are then made to determine the viscosity.

Viscometers for carrying out tests commercially depend upon the principle of allowing the liquid to flow from a vessel through a small hole, and finding the time necessary to collect a specified quantity. In Britain, the Redwood viscometer is the standard instrument, on the Continent the Engler is used,

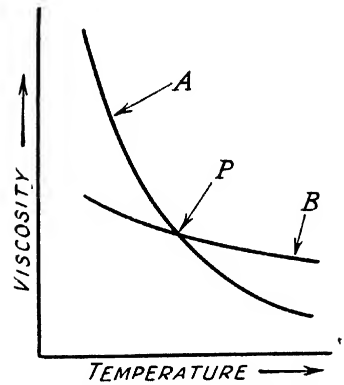


FIG. 5.—SHOWING THE VARIATION OF VISCOSITY WITH TEMPERATURE

The viscosity of the oil *A* falls rapidly with rise of temperature, but the oil *B* shows a much smaller change of viscosity over the same temperature range.

while in the U.S.A. the Saybolt viscometer is used. The Redwood viscometer is illustrated in Fig. 4. It consists of a cylindrical vessel with a small orifice in the centre of the base which can be closed by a ball valve.

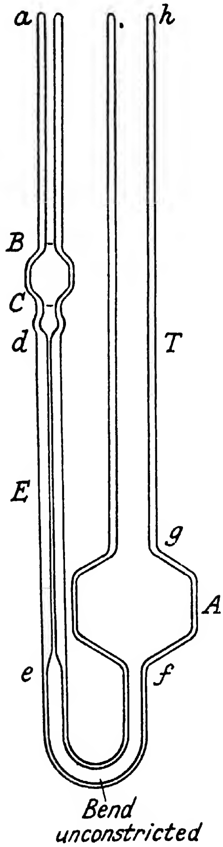
The vessel is surrounded by a jacket which can contain water or oil at a controlled temperature so as to maintain a constant temperature of the oil being tested. A copper tube, closed at the lower end, projects from the side of this jacket, so that the liquid in the jacket may be heated by a bunsen burner placed under the tube. Thermometers are fitted so that the temperature of the jacket and of the oil to be tested can be read. A stirrer is fitted to agitate the liquid in the jacket to ensure even jacket temperature.

The test is carried out as follows:

The central vessel is filled to the height of a hook gauge with oil at the required temperature. After ensuring that the temperatures are constant, the ball valve is lifted, and this allows the oil to flow through the orifice into the measuring vessel placed below. The viscosity of the oil in "Seconds Redwood" is given by the time in seconds for 50 c.c. of oil to flow through the orifice. For very viscous oils an instrument with a larger orifice, called the No. 2 Redwood viscometer, is used, the time of flow for this instrument being one-tenth of the time for the No. 1 Redwood viscometer.

In order to find the variation of viscosity with temperature, oil may be tested at a number of temperatures, and these are usually taken as 70° F., 100° F., 140° F., and 200° F. Higher or lower temperatures may be used for special oils. Graphs may be plotted to show the variation of viscosity with temperature. Fig. 5 shows curves plotted for two different oils. At low temperatures the oil *A* is very viscous. The viscosity of this oil falls rapidly with rise of temperature, so that at high temperatures it is very fluid. The oil *B* has a much flatter curve, showing a much smaller change of viscosity over the same temperature range. At the temperature *P* both oils have the same viscosity.

The term "viscosity index" is used to indicate the rate of change in the viscosity of an oil over a given range of temperature. An oil whose viscosity changes rapidly over a given range of temperature has a low viscosity index, while an oil with a small change of viscosity has a high viscosity index. This variation of viscosity with temperature is of importance when selecting oils for internal-combustion engines which may operate under conditions of widely varying temperatures. With these engines it is necessary to ensure easy starting up from cold, but at the same time to ensure that the viscosity at



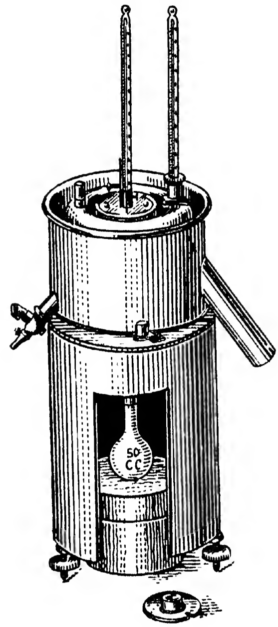


FIG. 3 (*left*).—A U-TUBE VISCOMETER

FIG. 4 (*above*).—THE REDWOOD VISCOMETER

The oil level is adjusted to the point *B*, and the oil is then allowed to flow back through the capillary tube under its own head. The time for the oil level to fall from *B* to *C* is taken, and from this the viscosity is calculated.

Another type of viscometer is used in which a sphere of specified diameter and density is allowed to fall through the liquid contained in a flask and the time is taken for the sphere to fall a specified distance. Calculations are then made to determine the viscosity.

Viscometers for carrying out tests commercially depend upon the principle of allowing the liquid to flow from a vessel through a small hole, and finding the time necessary to collect a specified quantity. In Britain, the Redwood viscometer is the standard instrument, on the Continent the Engler is used,

flashes is the open flash point. A lubricating oil with a high flash point is required for such machines as air compressors, where a high temperature is generated inside the compressor cylinder, due to the compression of the air, and where an explosion resulting from the ignition of the lubricating oil would have serious consequences.

If two oils having different flash points are mixed, the flash point of the resulting mixture may be found from

$$F = \frac{CW_1F_1 + W_2F_2}{CW_1 + W_2},$$

where:— C is a constant.
W_1 is the weight of oil No. 1.
W_2 is the weight of oil No. 2.
F_1 is the flash point of oil No. 1.
F_2 is the flash point of oil No. 2.

The constant C has a value of approximately 0·4 for American oils and 0·9 for Russian oils.

Fire Point

The fire point is the temperature at which the vapour of an oil commences to burn for a period of 5 secs. when a test flame is applied. The fire point of an oil is approximately 60° F. above the open flash point. The same apparatus as is used for the determination of the flash point may be used for the determination of the fire point.

Volatility

The readiness with which an oil loses weight by evaporation is its volatility. A fuel oil should be fairly volatile, so that it is readily converted into vapour for combustion. Lubricating oils should normally have a low volatility. To carry out the test a standard quantity of oil is placed in a vessel and heated at a specified temperature for a definite time. The oil remaining is weighed, and the loss of weight, expressed as a percentage of the original quantity, gives an indication of the volatility.

Cloud and Pour Points

It has been pointed out that as the temperature of an oil falls it becomes very viscous. If the temperature continues to fall, a point is reached where the oil becomes cloudy due to the formation of small crystals. This temperature is known as the "cloud point." If the temperature is further reduced, a point will be reached where the oil will only just flow when the vessel is tilted. Fig. 7 illustrates the apparatus used for the determination of the "cloud" and "pour" points. It consists of a cylindrical test jar tightly closed at the top by a cork,

through which fits the test thermometer. The test jar containing the oil is placed in a cooling bath, which extracts the heat from the test jar and oil until the cloud and pour points are reached.

Cold Test and Setting Point

An oil may be required to operate for a part of its time at extremely low temperatures under such conditions as may be experienced in aircraft flying at high altitudes or in pneumatic tools where the expansion of compressed air produces a chilling effect. It is essential that such an oil should remain fluid, and that there should be no danger of it setting due to the low temperature, and the oil must therefore have a low setting point.

The apparatus used for the determination of the setting point, that is, the point at which the oil ceases to flow when subjected to a small pressure, is illustrated in Fig. 8.

A sample of oil is placed in a specially designed U-tube which is placed in an air bath surrounded by a freezing mixture of ice and salt, or, for low temperatures, solid carbon dioxide. A thermometer is inserted in one leg of the U-tube, which can be connected to atmosphere or to the supply of compressed air stored in the Winchester quart bottle by means of the two-way cock. The other leg of the U-tube is connected to an indicator tube which contains a small quantity of coloured water to act as an index. When the oil is fluid the admission of low-pressure air to the U-tube will cause a movement of oil which will be revealed by the movement of the coloured water in the indicator tube. As the temperature of the oil falls, a point is reached at which no movement is detected when the air pressure is applied. This temperature is the setting point.

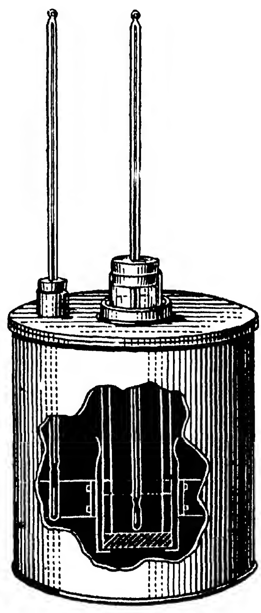


Fig. 7.—Apparatus for testing cloud and pour points

Carbon Residue Tests

These tests have been devised to determine the amount of carbon formed after a certain quantity of oil has been burned. The Conradson test is largely used in the U.S.A., but in Britain this has been largely replaced by the Ramsbottom test. Fig. 9 illustrates the Conradson apparatus, which consists of a porcelain crucible which is enclosed in a larger iron crucible fitted with a lid. This rests on sand contained in a still larger crucible, the whole of which is placed in a sheet-iron muffle that is heated from below. A weighed sample of oil is placed in the porcelain crucible

flashes is the open flash point. A lubricating oil with a high flash point is required for such machines as air compressors, where a high temperature is generated inside the compressor cylinder, due to the compression of the air, and where an explosion resulting from the ignition of the lubricating oil would have serious consequences.

If two oils having different flash points are mixed, the flash point of the resulting mixture may be found from

$$F = \frac{CW_1F_1 + W_2F_2}{CW_1 + W_2},$$

where:— C is a constant.
W_1 is the weight of oil No. 1.
W_2 is the weight of oil No. 2.
F_1 is the flash point of oil No. 1.
F_2 is the flash point of oil No. 2.

The constant C has a value of approximately 0·4 for American oils and 0·9 for Russian oils.

Fire Point

The fire point is the temperature at which the vapour of an oil commences to burn for a period of 5 secs. when a test flame is applied. The fire point of an oil is approximately 60° F. above the open flash point. The same apparatus as is used for the determination of the flash point may be used for the determination of the fire point.

Volatility

The readiness with which an oil loses weight by evaporation is its volatility. A fuel oil should be fairly volatile, so that it is readily converted into vapour for combustion. Lubricating oils should normally have a low volatility. To carry out the test a standard quantity of oil is placed in a vessel and heated at a specified temperature for a definite time. The oil remaining is weighed, and the loss of weight, expressed as a percentage of the original quantity, gives an indication of the volatility.

Cloud and Pour Points

It has been pointed out that as the temperature of an oil falls it becomes very viscous. If the temperature continues to fall, a point is reached where the oil becomes cloudy due to the formation of small crystals. This temperature is known as the "cloud point." If the temperature is further reduced, a point will be reached where the oil will only just flow when the vessel is tilted. Fig. 7 illustrates the apparatus used for the determination of the "cloud" and "pour" points. It consists of a cylindrical test jar tightly closed at the top by a cork,

measure the time for them to separate when the mixture is allowed to stand. Fig. 11 illustrates the apparatus used. Twenty c.c. of oil are placed in a graduated test-tube which is placed in the emulsification bath A that contains water.

The steam generator, B, is connected to the test-tube by means of glass and rubber tubing with a cork as shown. Thermometers D and F give the temperatures in the test-tube and bath. A separating bath A_1, also containing water, is connected to the steam flask in the same way. The water in A_1 is heated to 200–203° F. The water in A at the commencement is 67° F., and steam is admitted into the oil so that the oil temperature rises to between 190° F. and 195° F. The steam condenses on meeting the oil and emulsification takes place, which is continued until 40 c.c. of oil and water are contained in the test-tubes, the time for this being taken. The steam pipe is then removed, and the test-tube is transferred to the separating bath A_1. The time is then taken for 20 c.c. of oil to separate. This time is a measure of the demulsifying property of the oil, and provides the basis for the demulsification number.

The property of emulsification is desirable in some oils such as those which are used for coolants in machining operations. These soluble oils are mixed with water to form an emulsion, which is used on machine tools for cooling the tool and the metal, washing away cuttings, producing a smooth finish, and protecting the machined surface from corrosion and rust.

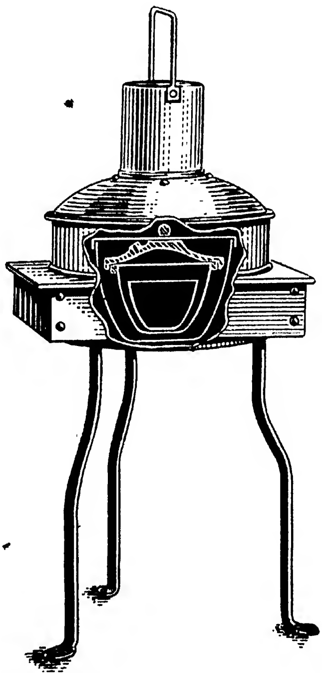


FIG. 9.—CONRADSON CARBON RESIDUE TEST APPARATUS

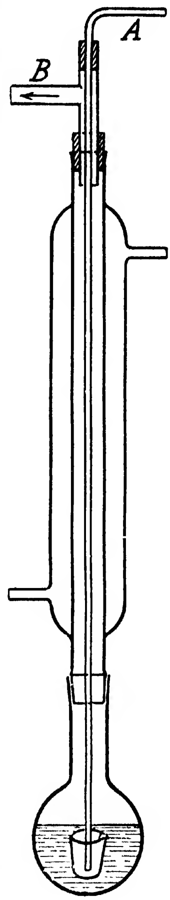


FIG. 10.—FLASK USED FOR SLUDGE TEST

Colour

The colour of oils varies from pale yellow

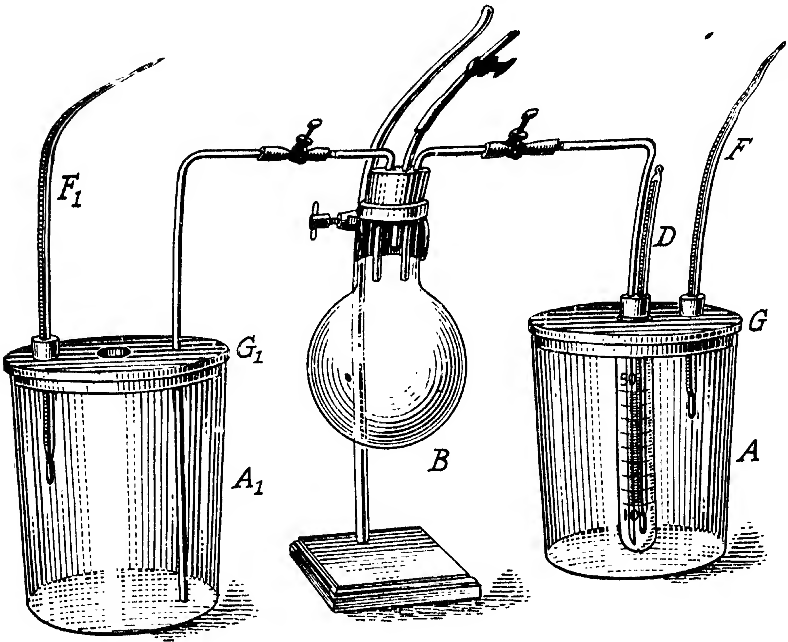


FIG. 11.—APPARATUS USED FOR EMULSIFICATION TEST

to red, brown, and almost black. This colour may be judged by transmitted or reflected light, that is, by light passing through the oil or reflected from its surface. The colour test by itself is not of great value, but it may help to distinguish between similar oils. An instrument known as a tintometer is used for carrying out colour tests, the principle being that of matching the colour of the oil against a standard colour on a coloured slide.

Dielectric Strength

When oil is used as an electrical insulator, it is necessary that its resistance to the passage of an electric current should be high. The importance of electrical transformers and switches at the present time necessitates that this property should be ascertainable. The test apparatus consists of a vessel fitted with electrodes across which a high-voltage spark is passed. These electrodes are submerged in the oil to be tested, and the voltage adjusted until arcing takes place between the electrodes.

Water Content

The initial precautions to be taken when testing oil for its water content is to ensure that the sample being tested is representative of the bulk of the

...al from which the sample has been ...ected. The sample is placed in a distillation flask which is connected by a tube with a condenser and a receiver. The contents of the flask are heated, and as the vapour rises from the distillation flask it is condensed and is collected in the receiver, which is graduated to indicate the amount collected. Distillation is continued at a specified rate until all the water has been collected in the receiver.

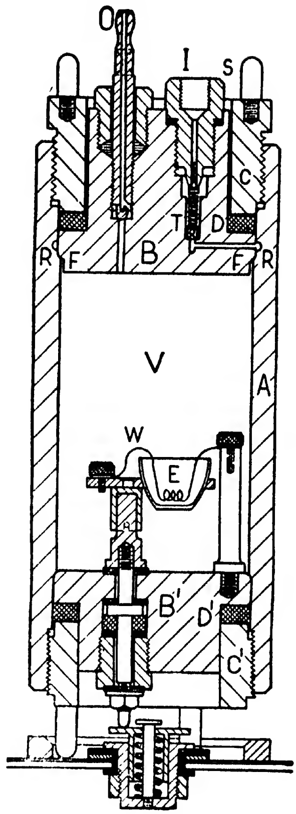


FIG. 12.—THE GRIFFIN-SUTTON BOMB CALORIMETER
(*Griffin & Tatlock, Ltd.*)

Diluent Tests

In internal-combustion engines the lubricating oil tends to become contaminated with the fuel oil and dilution of the lubricant takes place. The sample is tested by distillation so as to separate the mixture into its constituents.

Mechanical Testing

Numerous researches have been carried out on lubrication of bearings by many investigators. Many machines have been designed to determine the co-efficient of friction of an oil between a journal and a bearing, but many of these have been discarded, as they were only suitable for a relatively small range of operating conditions. The development of special gears, such as the hypoid in the automobile industry, have created problems in the lubrication field due to the high tooth pressures involved.

A variety of testing machines have been introduced to measure the strength of the lubricating film. Among these is the four-ball Extreme Pressure Lubricant Tester. This machine consists of three balls half an inch in diameter, fixed in an oil cup. A fourth ball, held in a vertical spindle which revolves at 1,450–1,500 r.p.m., is brought into contact with the three stationary balls. The amount of vertical pressure between the balls is adjustable, and the torque transmitted by the friction between the rotating ball and the three stationary balls is recorded on a chart. Different lubricants produce their own characteristic friction-time chart. The extreme pressure on the surface of the balls wears scars which are later measured under a microscope.

Calorific Value

In the case of fuel oils, the calorific value or the amount of heat which is produced by the combustion of the oil must be known. This calorific value is expressed in British Thermal Units (B.Th.U.) or Centigrade Heat Units (C.H.U.) per pound. The method of testing calorific value is to burn a weighed quantity of oil in a vessel called a calorimeter, the heat generated being absorbed by water surrounding the vessel. The rise in temperature of the water is measured by means of a very sensitive thermometer, and as the quantity of water is known, the heat generated by the combustion of the fuel may be calculated. A number of types of calorimeters are available, but the bomb type is used for accurate determination.

Fig. 12 illustrates a bomb calorimeter which consists of a thick-walled steel vessel with gastight joints at D and D'. A small quantity of fuel oil is weighed in the crucible E, and a short length of fuse wire W is fixed to a pair of terminals so that the wire touches the fuel. The bomb is assembled and filled with oxygen through the valve T to a pressure of about 30 atmospheres. The bomb is then immersed in a weighed quantity of water in a surrounding vessel which carries a sensitive thermometer also immersed in the water. The fuel is then ignited by passing an electric current through the fuse wire, and combustion of the fuel with the oxygen takes place inside the bomb. The heat generated is communicated through the walls of the bomb to the surrounding water, which therefore rises in temperature. Allowance must be made in the calculation for the water equivalent of the apparatus and for any loss of heat from the water jacket. The calorific value is calculated from

$$CV = W \times \frac{t}{w}$$

where:—

CV is a calorific value of the fuel in B.Th.U.s per pound.

W is the weight of water in pounds, including the water equivalent of the apparatus.

t is the temperature rise of the water in °F.

w is the weight of fuel burned in pounds.

Anyone intending to carry out oil tests on any scale should consult the relevant British Standard Specifications, and the publication of the Institute of Petroleum entitled *Standard Methods for Testing Petroleum and its Products*.

Illustrations have kindly been supplied by Messrs. Griffin & Tatlock, Ltd., and A. Gallenkamp & Co., Ltd.

R. S.

PULVERISED-COAL FIRING

THE system of firing certain types of boilers, such as water-tube boilers, and also metallurgical, forge, and other furnaces by means of pulverised coal has now been proved successful.

In this system, the coal is reduced to powder form and blown into a combustion chamber by a current of hot air. A further supply of air is blown in separately to ensure perfect combustion of the fuel.

Advantages of Pulverised-fuel Firing

Pulverised-fuel firing is not by any means new, although it is only within the last twenty years that it has been developed on a large scale. The advantages due to the use of pulverised fuel are:

(1) Higher furnace temperatures, owing to low excess air and good combustion conditions.

(2) Flexibility of the plant in operation, so that the fuel supply can be immediately, and, if required, automatically adjusted to the load, while still maintaining a high thermal efficiency.

(3) Rapidity in raising steam compared with mechanical or hand-fired coal furnaces.

(4) Elimination of stand-by losses.

(5) Reduced overall maintenance costs.

(6) Reduction in floor area for a given output.

(7) Improved labour conditions.

When Pulverised Fuel is Used

The use of pulverised fuel as an alternative method of firing steam boilers is largely governed by the appraisal of a number of factors, such as, size of boiler and type of load, steam cycle employed, local conditions.

(1) When the capacity of a boiler is of the order of, say, 200,000 lb. of steam per hour, involving the burning of coal at the rate of 25,000 lb. per hour, a very large mechanical stoker would be necessary, so much so that, in the opinion of many engineers, it would be far too unwieldy.

(2) In power-station practice, regenerative feed heating is increasingly made use of, which has the advantage of reducing condenser sizes and condensate loss. This results in the feed water to the boilers being practically at saturation temperature, with the result that convection-heating surface in the boilers or economisers is reduced to a minimum. To reduce the temperature of the products of combustion to a reasonable stack temperature, a large air heater must

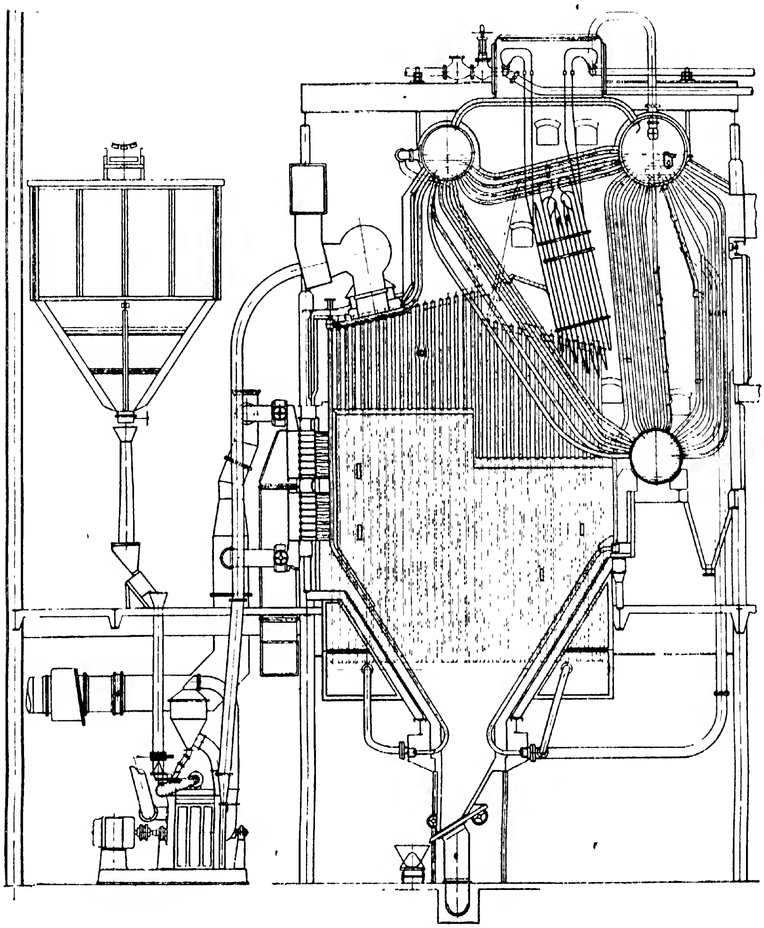


FIG. 1.—APPLICATION OF THE UNIT SYSTEM OF PULVERISED-COAL FIRING TO A SIMON-CARVES TWIN-FLOW COLD-FEED BOILER

be installed, with the result that the air for combustion is preheated to a high figure, of the order of 500° F. The use of stokers imposes a limit to preheated air temperature, usually taken as being 300–350° F., which does not apply with pulverised coal or oil.

(3) Local conditions, which have to be considered, may be numerous, or not. With many plants it frequently happens that the cheapest fuel available is dust,

which can only be fired in suspension. Alternatively, the coal may be unsuitable for stoker firing, either because its ash content may be too low, or its behaviour during the process of combustion is unsuitable.

Systems

In pulverised-coal firing a stream of finely divided dry coal, intimately associated with primary and secondary air, is projected into the furnace, where it ignites and burns in the same manner as atomised oil. Two systems are in use: the "Unit" system, in which coal is delivered and stored in a bunker, either direct or after a preliminary crushing, whence it is fed as required to a pulveriser which reduces it to combustion conditions, and the "Central" or storage system, where fuel is taken from a supply or main bunker to a drier where free moisture is extracted, and then to one or more pulverisers where it is reduced and then withdrawn to a pulverised fuel bunker for use as required. A combination of the two systems has been adopted in which each boiler has its own pulveriser, but a storage bin is provided for the pulverised fuel. This allows the pulverisers to work at the most economical load and not according to combustion conditions, and has the advantage of obtaining preheated air from the furnace, whilst the mill circuit can be vented back into the combustion chamber. A line drawing showing the application of the Unit system is given in Fig. 1, and a typical semi-direct installation will be seen in Fig. 2.

The Fuel

Practically any grade of coal can be effectively pulverised and efficiently consumed. Excessive ash and free moisture are both undesirable, as the former must be intercepted before reaching the chimney and the latter may have to be removed by drying.

In the case of the Central system the coal must be dried to less than about 3 per cent. moisture before pulverising and storing, otherwise there is a possibility of heating and spontaneous combustion in the storage bins, as well as the failure of the pulverised fuel to flow in outlet pipes and chutes.

With the Unit system a larger margin of water content will simply increase the power and cost of pulverising.

Ash

Coals with very high ash content can be used in pulverised form. Fuel containing 45 per cent. ash, formerly dumped as unburnable, has been pulverised and consumed, but, obviously, the higher the ash content, the greater will be the wastage of the wearing parts of the pulveriser. In some cases the cost of renewal of worn parts is outweighed by the greater saving in cost of fuel, particularly if it is available at a low price at the site and does not involve heavy transport cost. High ash content entails greater cost, owing to removal of ash, and may also cause erosion or wastage of combustion-chamber brickwork.

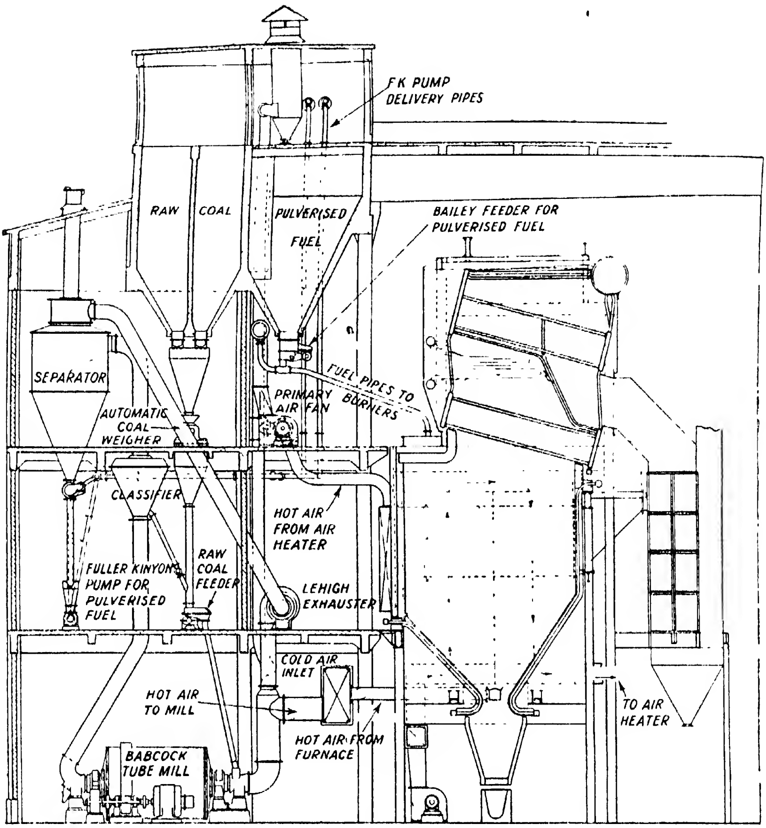


FIG. 2.—A TYPICAL SEMI-DIRECT INSTALLATION OF MESSRS. BABCOCK & WILCOX LTD.

Fineness of Grinding

Theoretically, the nearer the fuel approaches the gaseous state, the more intimate the mixture of air and fuel and the more rapid the combustion. In practice a uniform size based on actual experience with a particular grade of fuel is all that is required. Average good-quality bituminous coal is reduced to about 90 per cent. through 100 mesh and 85 per cent. through 200 mesh. Coal with low volatile content, such as anthracite and hard steam, requires finer and more uniform grinding than bituminous.

For cylindrical boilers the best results have been obtained with a high-grade bituminous coal having a calorific value of 13,000 B.Th.U.s or higher and a volatile content of about 25 per cent. The guarantee given with bulk supplies is

in accordance with the following specification: calorific value, not less than 14,000 B.Th.U.s; volatile content, not less than 30 per cent.; moisture as delivered, not exceeding 3 per cent.; ash content, not exceeding 4 per cent.; fineness, 98 per cent. through 100-mesh, 85 per cent. through 200-mesh B.S. sieve.

The purchase of bulk supplies appears to offer the best opportunity to the small consumer contemplating pulverised-fuel firing, not so much on account of difficulties in pulverising, as because of the need for an air drier to supply hot air for drying purposes, and a standby plant in case of overhaul and repair.

Ash Elimination

The greatest disadvantage and most difficult problem in connection with pulverised-fuel firing in cylindrical boilers is the elimination of ash deposits, and this is the main reason why the number of industrial boilers of this type equipped for pulverised fuel is still comparatively small.

A 30-ft. Lancashire boiler will consume about 40 tons of coal per week, and if the ash content is only 5 per cent., then 2 tons of ash must either be deposited in the flues or pass up the chimney. Unless steps are taken periodically to clear the flues, the evaporative power of the boiler will be seriously reduced. Not only must the ash be eliminated, but it must, in most cases, be prevented from passing up the chimney.

A satisfactory and efficient method has been evolved for dealing with ash deposits, and consists of a series of nozzles situated in the furnaces and flues. The first ash blowers are placed in the burners, the second some 14 ft. in the furnace flues, and subsequent nozzles are arranged in the centre and side flues. The operation of the blowers in proper sequence is obtained by means of a manifold valve, which sets each series of blowers in operation so that the ash from the first series is taken up by the second, and so on until the ash reaches and is discharged into a collecting chamber. During the blowing period the main flue dampers are closed, and the dampers leading to the collecting chamber are opened. When the ash is being blown into the chamber, a number of water-spray nozzles are brought into operation, and the ash is separated from the flue gases and deposited upon the floor of the chamber, and is washed away by the water from the spray nozzles to a sump adjoining the ash chamber.

A considerable amount of steam must be used in ash blowing, and figures do not appear to be available showing the amount, or its equivalent, and upon this the value of pulverised-fuel firing for small industrial units mainly depends.

Conveying the Fuel

Dry pulverised fuel can be transported without difficulty by air pressure, screw and worm conveyors, and pumps. It can also be conveyed pneumatically through flexible tubes horizontally, vertically, and round easy bends.

THE SIMON-CARVES PULVERISER

Type of Mill

The Simon-Carves hammer mill is of the horizontal-axis swinging-hammer type (see Fig. 3). The mill body comprises a heavy cast-iron cylindrical casing fitted internally with wear-resisting cast-iron liners—in sections for easy replacement—which are attached to the casing by means of setscrews. A steel shaft, which runs in roller bearings, carries a series of manganese steel hammers by means of different-sized hubs on the shaft. The swinging hammers are hinged to the hubs. On the same shaft is mounted externally to the mill a fan which draws a stream of air through the mill to carry the pulverised coal particles away when sufficiently ground. The casing of the mill is provided with a catch-pocket for the ejection of pyrites and other heavy foreign particles which might enter the mill with the coal.

Feeders

The coal from the overhead bunker is delivered into a small receiving hopper situated on top of the feeder, which consists of a rotating plate driven by a variable-speed A.C. motor. The coal, which falls on to the centre plate through an adjustable sleeve, is swept off into the mill inlet by means of an adjustable plough or knife.

The control of the feed is normally performed by the variable-speed A.C. commutator motor. Adjustments can also be made by changing the height of the sleeve or the position of the plough.

Classifier

In order to prevent large particles of fuel reaching the burners, the mills are provided with a Simon-Carves static-type classifier. This consists of an inner and outer cone constructed of steel plate. The fuel and primary air enter at the side of the outer cone, the separation occurring at the periphery of the inner cone. The oversize particles fall to the bottom of the separator and are returned to the mill through a rotary sealing valve driven by a ½-h.p. motor, which allows the coal particles to pass but which prevents the air from by-passing the mill. The classifier is on the suction side of the fan; thus erosion of the fan impeller due to passing large particles is prevented.

The degree of separation may be regulated by adjusting the position of the inner cone. By raising this cone the annular space for the passage of air and fuel is decreased, the air velocity is increased, and larger fuel particles are entrained and pass to the burner. The converse operation reduces the size of the particles, so that a ready means of adjusting for wear of the beaters is provided.

Fixed magnetic separators can be arranged below the coal-feeding table or in the feed chute, which prevent any small tramp iron reaching the pulverisers.

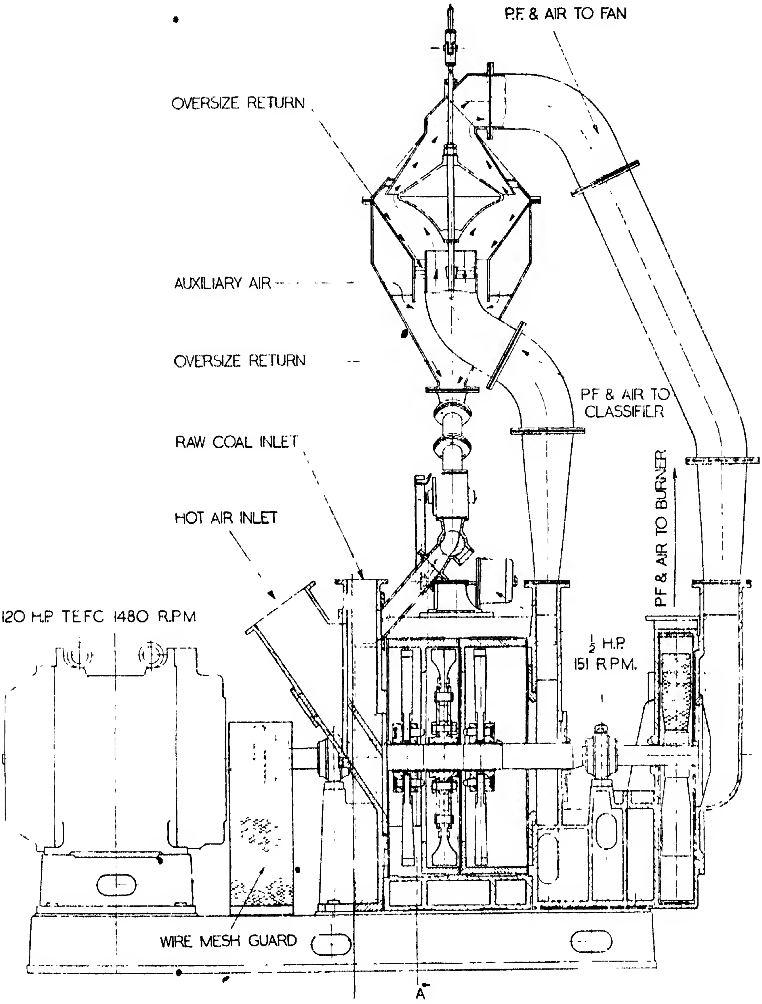


FIG. 3.—ARRANGEMENT OF SIMON-CARVES UNIT PULVERISER

Moisture in Fuel

The moisture content of the raw fuel has a large effect on the output and the fineness of grinding in the mill. It is therefore advisable with high-moisture coals to have hot air available. By this means it is possible to handle coals having up to 20 per cent. moisture. The quantity of air required for the carrying

THE SIMON-CARVES PULVERISER

Type of Mill

The Simon-Carves hammer mill is of the horizontal-axis swinging-hammer type (see Fig. 3). The mill body comprises a heavy cast-iron cylindrical casing fitted internally with wear-resisting cast-iron liners—in sections for easy replacement—which are attached to the casing by means of setscrews. A steel shaft, which runs in roller bearings, carries a series of manganese steel hammers by means of different-sized hubs on the shaft. The swinging hammers are hinged to the hubs. On the same shaft is mounted externally to the mill a fan which draws a stream of air through the mill to carry the pulverised coal particles away when sufficiently ground. The casing of the mill is provided with a catch-pocket for the ejection of pyrites and other heavy foreign particles which might enter the mill with the coal.

Feeders

The coal from the overhead bunker is delivered into a small receiving hopper situated on top of the feeder, which consists of a rotating plate driven by a variable-speed A.C. motor. The coal, which falls on to the centre plate through an adjustable sleeve, is swept off into the mill inlet by means of an adjustable plough or knife.

The control of the feed is normally performed by the variable-speed A.C. commutator motor. Adjustments can also be made by changing the height of the sleeve or the position of the plough.

Classifier

In order to prevent large particles of fuel reaching the burners, the mills are provided with a Simon-Carves static-type classifier. This consists of an inner and outer cone constructed of steel plate. The fuel and primary air enter at the side of the outer cone, the separation occurring at the periphery of the inner cone. The oversize particles fall to the bottom of the separator and are returned to the mill through a rotary sealing valve driven by a ½-h.p. motor, which allows the coal particles to pass but which prevents the air from by-passing the mill. The classifier is on the suction side of the fan; thus erosion of the fan impeller due to passing large particles is prevented.

The degree of separation may be regulated by adjusting the position of the inner cone. By raising this cone the annular space for the passage of air and fuel is decreased, the air velocity is increased, and larger fuel particles are entrained and pass to the burner. The converse operation reduces the size of the particles, so that a ready means of adjusting for wear of the beaters is provided.

Fixed magnetic separators can be arranged below the coal-feeding table or in the feed chute, which prevent any small tramp iron reaching the pulverisers.

mass of coal consisting of superfines, fines, and coarse particles. The coal stream strikes the deflector plate *B* and is spread out in the form of a disc of coal-dust.

The primary air fan, which draws through the suction pipe *D*, picks up the fine particles from the coal-dust, and the heavier and coarser particles fall down and return through the passages *E* to the beater chamber for further pulverisation. The degree of fineness is controlled by the position of the deflector plate, and this position can be varied by means of the rods *C*.

The separator is cone-shaped, with a short cylindrical section near the top, and it is at this point where the volume is greatest that effective separation takes place. The deflector can be raised or lowered until the point is reached where the desired fineness is obtained, and this is usually where the upward pull of the primary fan through *D* is balanced by the downward pull of the beater disc through the return passages *E*.

Pulverising Chamber

The pulverising chamber is made of semi-steel, and is lined throughout with renewable hard-steel liners. These liners are interchangeable where possible, and of such a size that they can be easily handled.

The pulverising disc is made of mild steel, and is secured on the main shaft by driving pins and locked by a nut. The door of the chamber is kept shut by hinged bolts, and provided with wing nuts so as to ensure rapid examination.

Motor Drive

The driving motor may be of the pipe-ventilated or totally enclosed type and suitable for either A.C. or D.C.

THE KENNEDY PULVERISER

The pulveriser shown in Fig. 6 is an impact ball mill of slow speed. It is constructed in various diameters from 24 in. to 96 in. and in lengths desired for producing a given tonnage. The heads of these mills have trunnions cast integral. A steel shell with flanges on either end is bolted to the heads, and corrugated liners are keyed to the inside, with the worm drive mounted on the discharge end trunnion. The barrel of the mill is charged with forged-steel balls, the balls being supplied through the feed end while the mill is in operation. The wormgear is mounted on a flange cast integral with the trunnion and the worm is mounted in roller bearings and is directly connected to a motor mounted on the

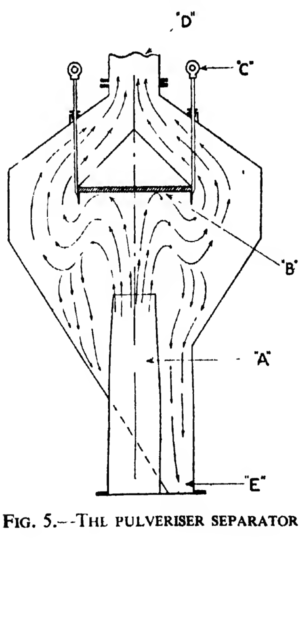


FIG. 5.—THE PULVERISER SEPARATOR

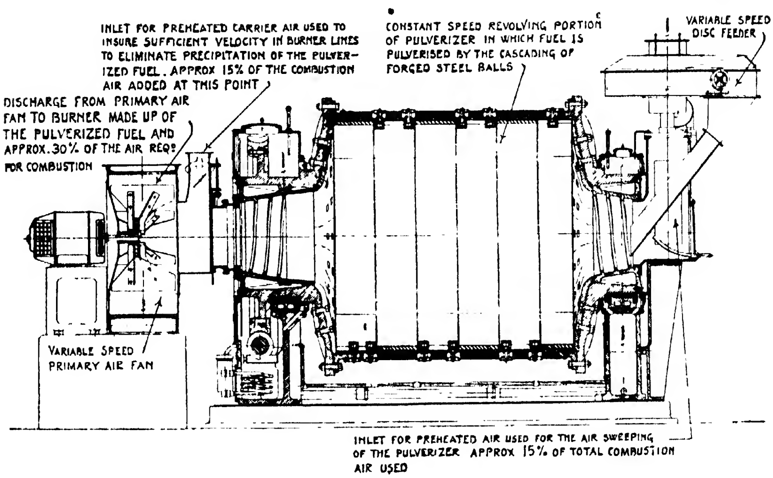


FIG. 6.—CROSS-SECTION OF A KENNEDY PULVERISER
(*Sheepbridge Engineering, Ltd.*)

same bed plate as the mill bearings, thus making a self-contained unit.

Material is fed into the mill in desired quantities by a disc feeder, usually driven by a variable-speed motor. The mill is rotated at a speed which causes the balls to be carried round by centrifugal force to a point where they are thrown down on to the material to be ground.

The cost of maintenance when crushing bituminous coal is normally about 8*d.* per ton, and when crushing anthracite about 1*s.* 2*d.* per ton.

The mill has no mechanical discharge, but depends upon air passing through to float out the fine particles. The fan connected to the discharge-end trunnion, driven by a variable-speed motor, draws air through the mill. The amount of air, and thus the velocity, may be varied by the fan speed and also by the inlet and outlet dampers; thus only the impalpable powder is removed by the mill.

THE ATRITOR PULVERISER

The coal, which should be limited in size to $\frac{3}{4}$-in. cubes, is fed into the hopper on the machine and descends on to a horizontal rotating disc. The spreading of the coal is controlled by a sleeve. which is adjustable for height, and the feed of the coal to the machine is regulated by a knife, which scrapes the coal off the disc. The position of the knife is controlled by handwheel.

The feeder is designed to obtain fine regulation of the coal, and is capable of dealing with any normal coal containing up to 15 per cent. moisture content. From the feeder the coal descends to the separator, where metals and foreign substances heavier than coal are rejected.

Passing from the metal separator, the raw coal enters the pulverising compartment at the centre, where it is subjected to the disintegrating effect caused by a number of hammers which beat the incoming coal against the liner ring that surrounds them.

After this treatment the coal is much reduced in size and can be carried over the periphery of the rotor into the attrition zone by the conveying air. In this zone the movement of the particles is most complex. They are carried inward to the centre and outward again to the periphery and back again until the particles are reduced in size and mass. This turbulence is created by a series of impellers mounted on the rotor, which set up eddies and vortices that rub the coal particles together. Fixed pegs or interrupters facilitate and accelerate the creation of these vortices and also prevent the gyration of the coal-dust with the rotor.

When the coal particles have become superfine, they are gradually drawn out towards the centre and intercepted by a device termed a rejector. This prevents any oversize particles being drawn by the fan from the pulverising zone and delivered to the burner.

The drying of the coal is effected by a supply of hot air or flue gases. The general construction of the pulveriser will be seen in Fig. 7, which gives a section through the machine.

The line-drawing, Fig. 8, shows the general layout of a Herbert-Brett drop-forging furnace and waste-heat boiler equipped for pulverised-fuel firing.

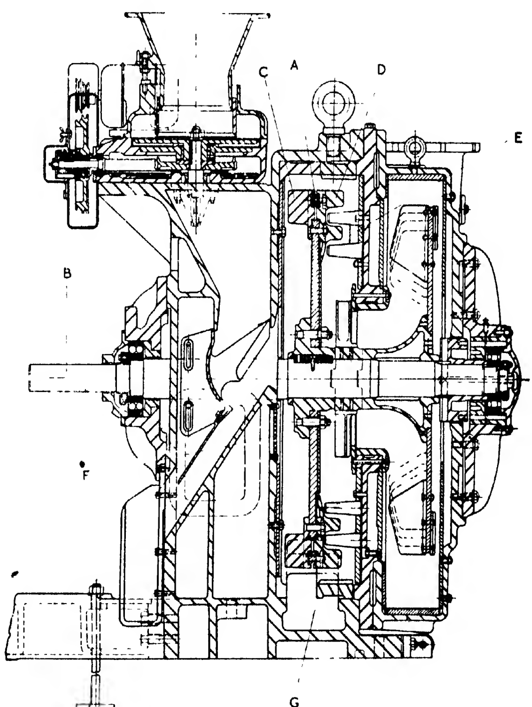


Fig. 7.—Section through an Atritor pulveriser
(*Alfred Herbert, Ltd.*)

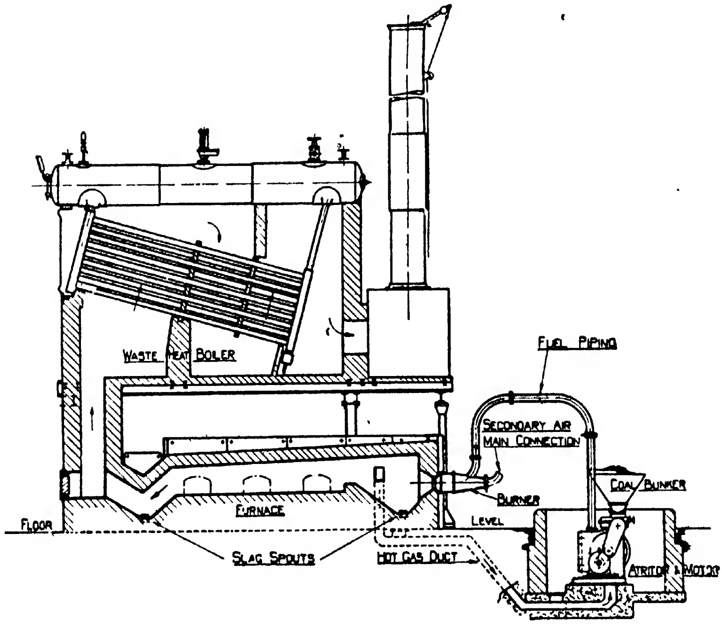


FIG. 8.—AN ATRITOR INSTALLATION APPLIED TO A HERBERT-BRETT DROP-FORGE FURNACE AND WASTE-HEAT BOILER

THE BABCOCK & WILCOX TUBE MILL

An outline diagram illustrating the main features of the Babcock & Wilcox tube mill is given in Fig. 9. The fuel is introduced into the shell of the mill by means of a coal feeder, usually of the rotary table type. This is worm driven either by means of a chain from the mill drive or by an independent 2-h.p. motor. The shell into which the raw coal is fed is constructed of mild-steel plate rolled to the form of a "tube" or cylinder 6 ft. or more in diameter, and of a suitable length to give the maximum commercial fineness of the product.

The liners of the shell are constructed of chrome steel, which has hardness and abrasion resistance without a tendency to spread, and are of special shape designed to induce agitation of the ball charge.

The ball charge consists of a graded quantity of hard-steel balls in four sizes, apportioned to suit the size reduction of the coal to be pulverised. The actual pulverising is effected by this ball charge when the mill slowly rotates. The balls cascade with the entrained coal, which is thus subjected to both impact and abrasion on all sides.

Around the outer circumference of the inlet end of the mill is a gear ring, by which the mill is driven. The pinion which engages with this girth gear is directly

Passing from the metal separator, the raw coal enters the pulverising compartment at the centre, where it is subjected to the disintegrating effect caused by a number of hammers which beat the incoming coal against the liner ring that surrounds them.

After this treatment the coal is much reduced in size and can be carried over the periphery of the rotor into the attrition zone by the conveying air. In this zone the movement of the particles is most complex. They are carried inward to the centre and outward again to the periphery and back again until the particles are reduced in size and mass. This turbulence is created by a series of impellers mounted on the rotor, which set up eddies and vortices that rub the coal particles together. Fixed pegs or interrupters facilitate and accelerate the creation of these vortices and also prevent the gyration of the coal-dust with the rotor.

When the coal particles have become superfine, they are gradually drawn out towards the centre and intercepted by a device termed a rejector. This prevents any oversize particles being drawn by the fan from the pulverising zone and delivered to the burner.

The drying of the coal is effected by a supply of hot air or flue gases. The general construction of the pulveriser will be seen in Fig. 7, which gives a section through the machine.

The line-drawing, Fig. 8, shows the general layout of a Herbert-Brett drop-forging furnace and waste-heat boiler equipped for pulverised-fuel firing.

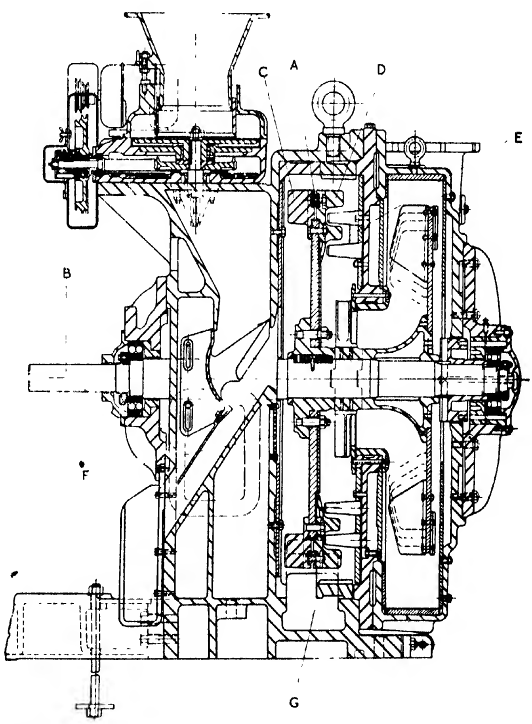


FIG. 7.—SECTION THROUGH AN ATRITOR PULVERISER
(*Alfred Herbert, Ltd.*)

According to the system of firing to be adopted, the product from the classifier passes to the collecting cyclone in the case of the storage and semi-direct system and immediately to the burner in the unit system.

This type of mill is constructed in six standard sizes with normal capacities of 5–15 tons per hour.

THE BABCOCK & WILCOX TYPE "E" PULVERISER

The type "E" pulveriser consists essentially of two horizontal grinding rings, between which is arranged a row of balls. The coal is fed into the middle of the top ring, the pulverising action taking place as it finds its way, aided by centrifugal force, between the rotating balls, where it drops over the edge of the bottom ring into an upward current of hot air, which at this point has a comparatively high velocity. Separation is effected by the rotating classifier assisted by the drop in velocity of the coal and air in the upper portion of the pulveriser. The fines are carried on by the air to the burners, whilst the oversize particles are returned for further grinding.

The automatic feeder control operates so that the output of material is governed by the quantity of air fed to the pulveriser and maintains the correct relation of coal to air over the entire operating range. At reduced outputs there will be a lower air velocity through the mill and, consequently, high fineness, which is most desirable in order to maintain good furnace conditions, particularly when firing coal with a low volatile content.

Owing to the absence of any considerable coal-storage capacity, the pulveriser is extremely responsive, one test showing an increased and maintained steam flow from 100,000 lb. per hour to 276,000 lb. per hour in less than three minutes.

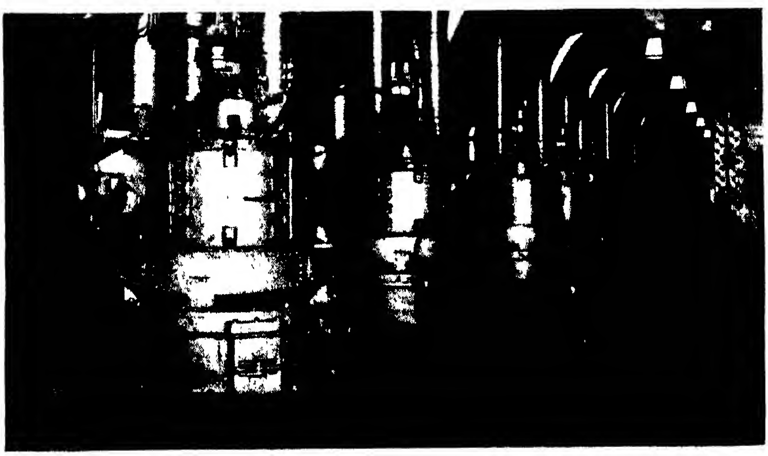


FIG. 10.—TYPE "E" MILLS INSTALLED AT MEAFORD POWER-STATION
(*Babcock & Wilcox, Ltd.*)

Grinding Elements

The grinding elements (Fig. 12) consist of one row of forged-steel balls between the rotating bottom grinding ring and the stationary top grinding ring. The balls are propelled by the rotating bottom grinding ring which is driven from the main driving shaft. Grinding pressure between the balls and rings is applied and kept uniform by steel springs in the top section arranged for independent external adjustment.

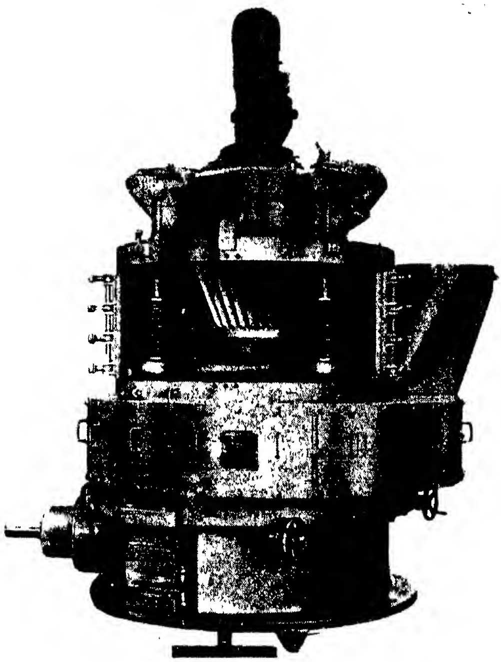


FIG. 11.—EXTERNAL VIEW OF MILL, WITH ACCESS DOOR REMOVED (*Babcock & Wilcox, Ltd.*)

Air Supply

This is provided by a fan operating on the clean air side of the mill. It is possible, therefore, to utilise a high-efficiency fan, of steel-plate construction, as the impeller is not subjected to erosion by coal particles as occurs when the fan is situated between the pulveriser and burners.

Raw-coal Feeder

The pulveriser is provided with a table feeder of the slow-speed rotating-disc type, designed to give a regular feed over the full range of operation. The rotating table and scraper gear are enclosed in a mild-steel housing and special provision is made in the construction of the driving mechanism to prevent the infiltration of coal dust. Operation of the rotating table is provided for by means of a two-speed motor.

Automatic Feeder Controller

The control system illustrated in Fig. 13 automatically maintains a constant ratio between the rate of air flow and the resistance to the flow of air through the pulveriser. Changes in the rate of coal fed to the burners are made

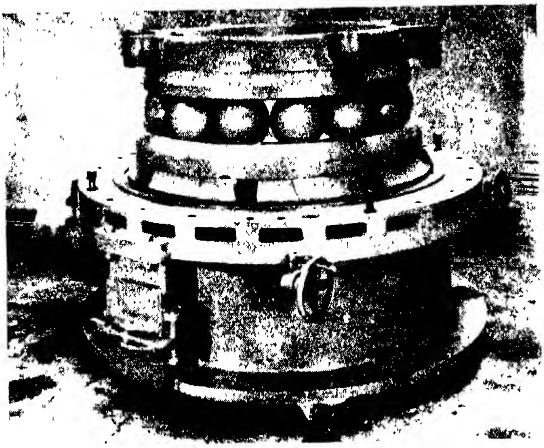


FIG. 12.—GRINDING ELEMENT OF MILL, SHOWING PYRITES TRAP DOOR (*Babcock & Wilcox, Ltd.*)

by merely adjusting the damper in the hot-air pipe, and a uniform mixture of pulverised coal and air in the correct proportions for efficient combustion is always delivered to the burners.

The mechanism of this control consists of two diaphragms working in opposition to each other, one actuated by the air-pressure differential across the orifice plate, measuring the air flow through the pulveriser, and the other actuated by the drop in air pressure through the pulveriser, which varies with the quantity of coal in the pulveriser. Any unbalanced relation between the two diaphragms changes the position of the contact points.

The position of the contact points determines the speed of the feeder motor.

FURNACES AND BURNERS

Three different designs of furnaces for pulverised-fuel firing in water-tube boilers are now recognised:

(1) Flat-bottom furnaces with rake-out removal of the solid ash.

(2) Hopper-bottom furnaces from which the ash is discharged, either in solid or in liquid form.

(3) Slag-tap furnaces in which the ash is removed in liquid form only.

Burners

The Simon-Carves burner is shown in Fig. 14. It is mounted on the top of the combustion chamber, and consists of a rectangular fire-brick nozzle connected to a mixing chamber, designed to give a sharp change of direction from horizontal to vertical, and thus create turbulent conditions.

Flat-flame Burner

The flat-flame burner illustrated in Fig. 15 is used for billet heating and similar furnaces. The fuel is ejected from a flat nozzle into a chamber into which

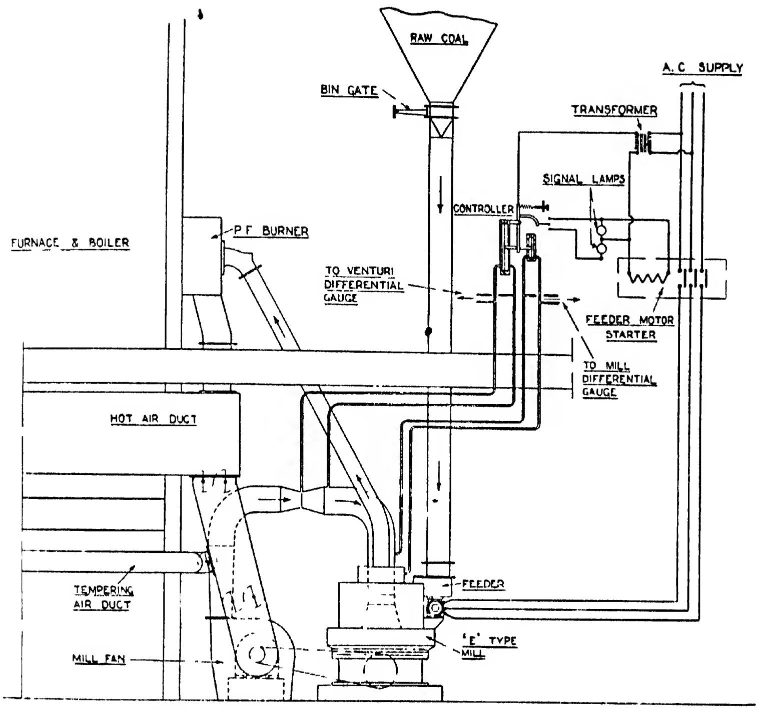


FIG. 13.—ARRANGEMENT OF BAILEY AUTOMATIC MILL CONTROL
(*Babcock & Wilcox, Ltd.*)

secondary air is induced by ejection action. The mixture of fuel and air is then passed through a further nozzle and into the furnace, tertiary air being added at this point from a forced-draught fan.

Dispersive Burner

The function of any burner is to mix the combustible so intimately with the air necessary for combustion that complete combustion is obtained with the nearest approach to the theoretical quantity of air. In practice there are so many varying conditions in industrial furnaces necessitating short flames at the highest possible temperature and extremely long flames with a medium temperature, that it is not possible to use the same type of burner for each application.

Where maximum flame temperatures are required, the dispersive burner shown in Fig. 16 has been developed. The primary air and fuel are delivered through the centre tube and the secondary air at 1½–2 in. of water-gauge pressure for the completion of combustion through the outer tube. When the primary

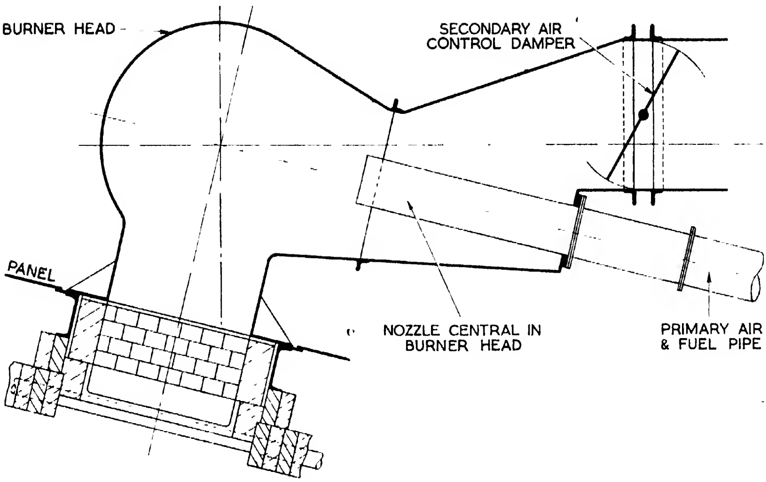


FIG. 14.—THE SIMON-CARVES BURNER

air and coal emerges, its direction is abruptly changed and its continuity broken, so that it is intimately associated with the surrounding or secondary air.

The Woodeson Burner

The primary object of this burner is intimately to mix the pulverised fuel and the air for combustion inside the burner before it is discharged into the combustion chamber and thereby ensure a steady and intense flame.

There are two louvres for controlling the air entering the burner, the main air louvre *A* and the secondary air louvre *B* (Fig. 17).

The main air flows through the burner in the same direction as the fuel and primary air, but the auxiliary air passes through the small tubes and pierces the fuel and air stream at right angles, thus causing intense turbulence and mixing of fuel and air.

All the air and fuel is further mixed and given a rotary motion through the adjustable deflector *C* before leaving the burner.

The length of the flame can be controlled by adjusting the main and auxiliary air louvres. For instance, if a long flame is required, the majority of the air should pass through the main air louvre, and if a short flame the greater portion of the air should pass through the auxiliary air louvre, as this air, which enters at right angles, retards the flow through the burner.

For lighting-up purposes an oil torch can be passed through the sight hole *D* until it projects in front of the burner.

This design of burner (Fig. 17) is made in sizes firing from 30 lb. to $2\frac{1}{2}$ tons per hour and has been fitted to industrial furnaces, water-tube boilers, cylindrical

marine boilers, and Lancashire boilers.

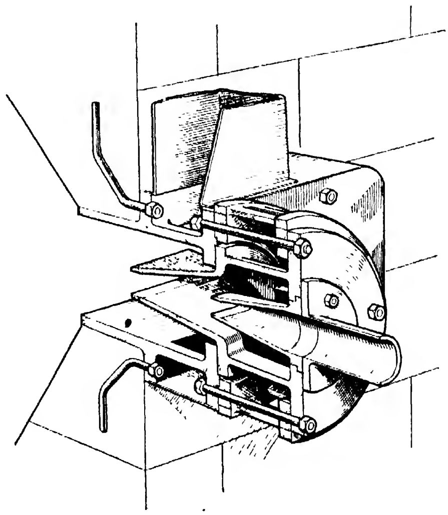


FIG. 15.—FLAT-FLAME BURNER (*Sheepbridge Engineering, Ltd.*)

The Lodi Burner

In the burner shown in Figs. 18 and 19, the coal and primary air enter through the central pipe and are spread by the deflected vanes, which impart a swirling motion. The stream of secondary air passes through the annular space surrounding the pipe, and is caused to meet the primary air and coal in such a manner as to produce an intimate mixture at the burner tip and a short flame of circular cross-section.

A minimum of bare metal is exposed to direct radiation, the only part being the deflector vanes, which are cooled by the incoming primary air.

The secondary air for combustion may be induced, but preferably is supplied under pressure, and may be at atmospheric temperature or preheated.

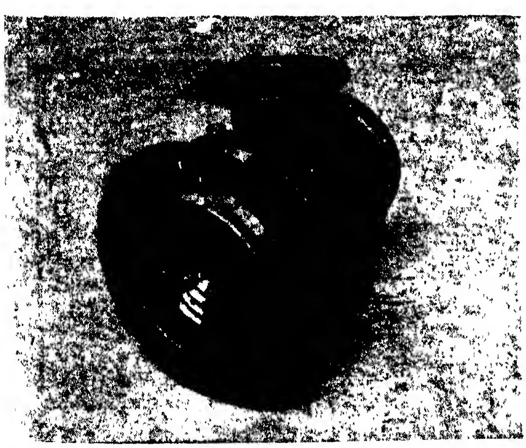


FIG. 16.—WATER-COOLED DISPERSIVE BURNER (*Alfred Herbert, Ltd.*)

This type of burner is suitable for storage systems or direct-fired application, and is built in sizes up to 80 million B.Th.U.s per hour liberating capacity.

The Calumet Burner

This is a highly turbulent burner for firing through a water-cooled furnace wall (Fig. 20), thus making possible a completely water-cooled furnace.

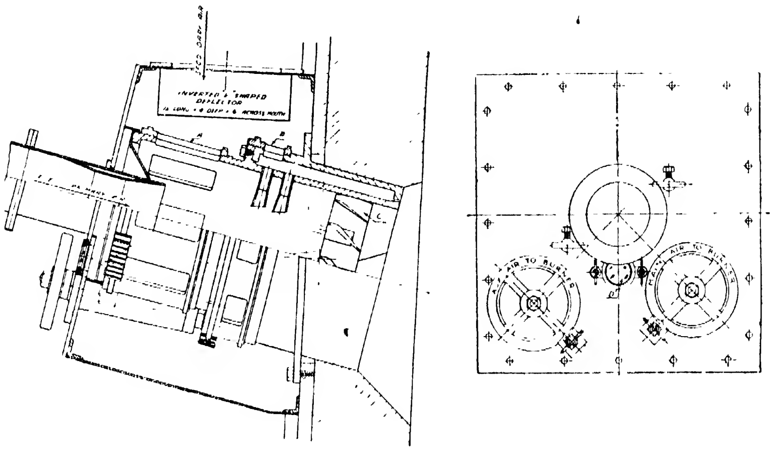
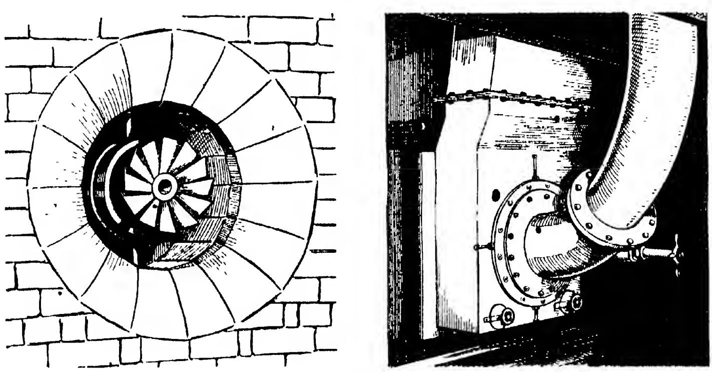


FIG. 17—THE WOODESON BURNER
(Clarke, Chapman & Co., Ltd.)

The coal-laden primary air is admitted through a long narrow vertical slot (Fig. 20), on either side of which are disposed at an angle the staggered secondary air ports. An intimate mixture of coal and air is obtained at the mouth of the burner, resulting in a short turbulent flame of rectangular cross-section, which enables full use to be made of the combustion space.

The burner is completely water cooled by being tightly clamped to the water tubes, which allows burners to be shut down on partial boiler outputs without fear of damage from overheating.



Interior view. Exterior view.

FIG. 18.—THE LODI BURNER INSTALLED ON A BABCOCK & WILCOX WATER-TUBE BOILER

The Calumet burner is manufactured by Babcock & Wilcox, Ltd., in sizes up to a heat-liberating capacity of 120 million B.Th.U.s per hour, and is therefore suitable for the largest units with either the storage or the direct-fired system.

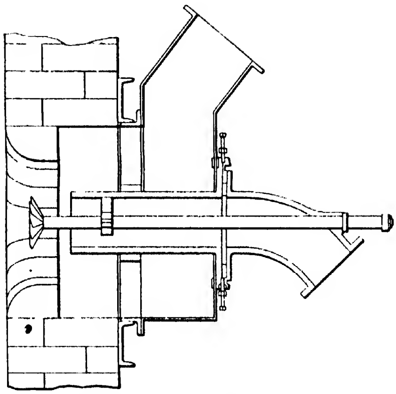


FIG. 19.—SECTION THROUGH LODI BURNER

Lighting-up Arrangements

Pulverised-fuel firing may be started by first providing a wood or coal fire at the bottom of the combustion chamber or with the aid of a paraffin or gas torch capable of igniting the stream of coal in 5–10 seconds. The flame should then become stable in 30–40 seconds.

On large installations a small motor-driven air compressor with cylindrical receiver for fixing in a convenient position in the boiler-house is often provided to operate paraffin burners placed in the lighting-up doors of the combustion chamber. The oil is siphoned from a portable drum, atomised, and projected into the combustion chamber by means of the compressed air. A wire basket containing wool or tow soaked in paraffin is ignited in order to start the paraffin torches.

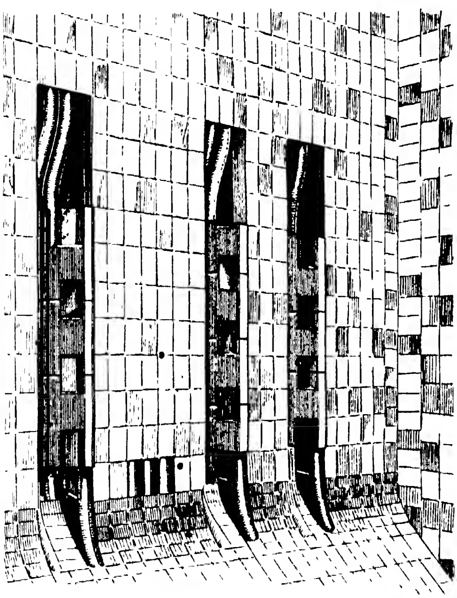


FIG. 20.—VIEW FROM INSIDE FURNACE SHOWING CALUMET BURNERS DURING ERECTION, WATER-COOLED FRONT AND SIDE WALLS, AND HOPPER BOTTOM

If a rich mixture of pulverised fuel and air is used with a low air velocity, ignition should take place in a few seconds; but should this fail, the combustion chamber must be cleared of gases before a second attempt is made.

A short stoppage in

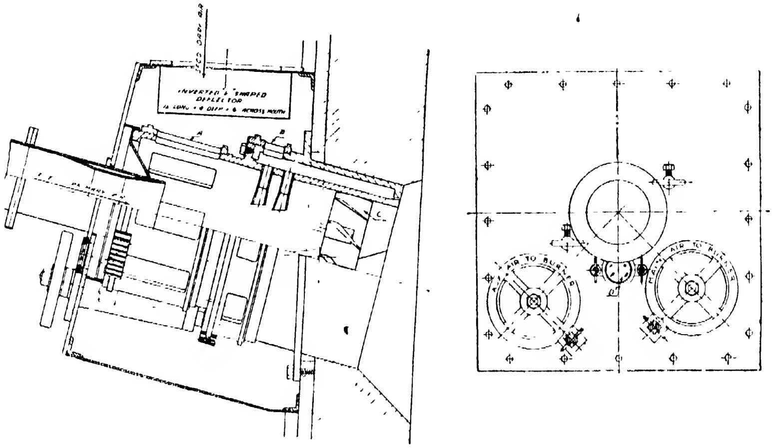
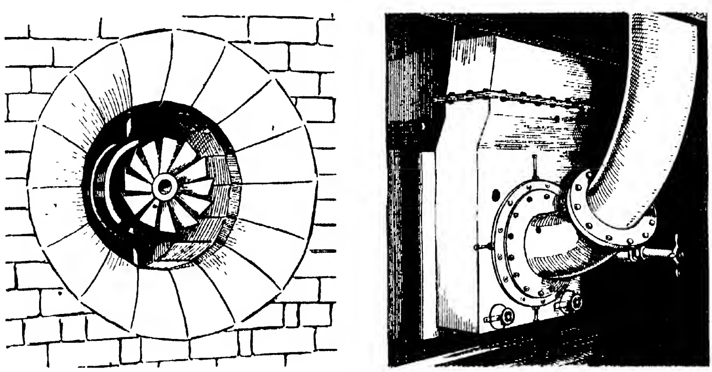


FIG. 17.—THE WOODESON BURNER
(*Clarke, Chapman & Co., Ltd.*)

The coal-laden primary air is admitted through a long narrow vertical slot (Fig. 20), on either side of which are disposed at an angle the staggered secondary air ports. An intimate mixture of coal and air is obtained at the mouth of the burner, resulting in a short turbulent flame of rectangular cross-section, which enables full use to be made of the combustion space.

The burner is completely water cooled by being tightly clamped to the water tubes, which allows burners to be shut down on partial boiler outputs without fear of damage from overheating.



Interior view. Exterior view.

FIG. 18.—THE LODI BURNER INSTALLED ON A BABCOCK & WILCOX WATER-TUBE BOILER

TESTING SOLID FUELS

THE rising cost of fuel and labour has drawn attention to many aspects of economical fuel consumption that have hitherto been neglected even in fairly large concerns, and greatly increased consideration is now being given to problems affecting economic efficiency. Unfortunately, the thermal efficiency of fuel consumption and combustion can only be arrived at in a very roundabout manner, and this is probably the reason why so many fuel-consuming plants are being run at a cost well above what could be reasonably expected.

The most useful data upon which to assess the efficiency of fuel burning in steam-generating and other plants are the figures representing the calorific value of the fuel and the percentages of carbon dioxide and carbon monoxide in the waste gases. To obtain the former with great accuracy calls for considerable expert experience and expensive instruments, but extremely useful results can be obtained by any intelligent person sufficiently interested in the subject to give the matter a little time and consideration, and no matter whether the fuel consumer or the owner of a steam-generating plant adopts fuel and gas analysis or not, it is most desirable that he should be acquainted with the reasons for such analyses and the methods that can be adopted to obtain satisfactory and consistent results.

Fuel and the Calorimeter

The calorific value of a fuel is the quantity of heat a unit weight will give out when it is completely burnt and the products of combustion are cooled to the original temperature. This is usually expressed in British Thermal Units per pound, and may vary between 10,000 and 15,000 B.Th.U.s in the case of coal.

To obtain the calorific value, a calorimeter is used in which a prepared sample of the fuel is burned in a vessel containing water, and the comparison of the original and final temperatures of the water gives the calorific value. Many types of calorimeters are in use, one of the simplest and least expensive being the "Ronald Wild" bomb type illustrated in Fig. 1. This consists of a combustion chamber with a screwed cap suspended from a cover by a tube furnished with a valve. The water vessel is of plated copper, and is surrounded by an outer casing, the annular space between the two vessels forming an air jacket which prevents radiation and absorption. An agitating paddle is provided for working in the water space and is manipulated from the outside of the cover. A Fahrenheit thermometer graduated 40°–100° is provided, the scale being divided into tenths of a degree for easy reading to one-twentieth of a degree.

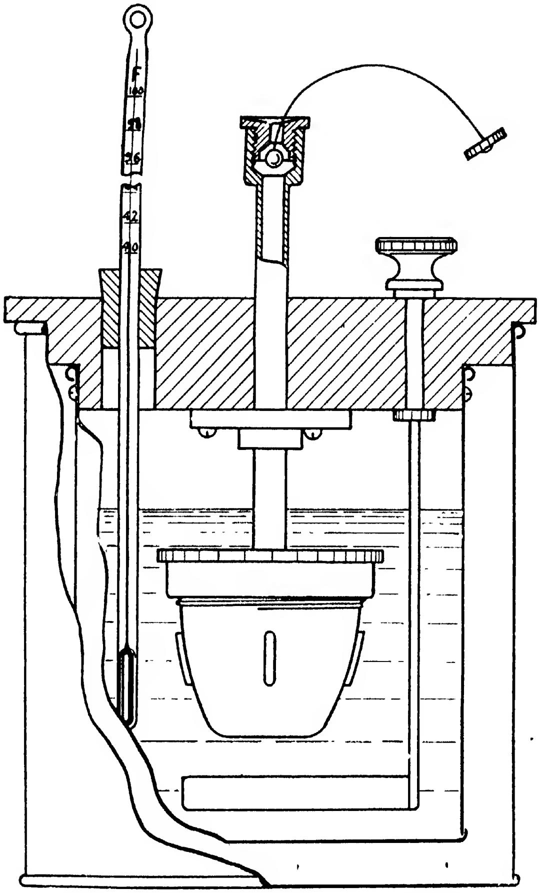


FIG. 1.—THE "RONALD WILD" FUEL CALORIMETER

Operation of Calorimeter

The operation of the calorimeter is as follows: the sample of coal to be tested is carefully ground to pass through a sixty-mesh sieve or smaller and dried in an air oven at about 220° F. for one hour or until completely dry. Of the sample 0·73 grm. is weighed, mixed with 12–14 grm. of sodium peroxide, and transferred to the combustion chamber and the cap firmly secured. Water equivalent to 1,000 c.c., less the certified water value of the calorimeter, is poured into the water vessel. The combustion chamber is placed in the water vessel and the thermometer inserted in the hole in the cover. The paddle is then agitated and the temperature of the water taken. A small piece of red-hot nickel wire is dropped down the hole communicating with the combustion chamber and the ball valve closed. The paddle is kept in motion for four to five minutes, and the maximum temperature of the water taken. The difference between the initial temperature and the final temperature multiplied by 1,000 is the gross calorific value of the fuel in British Thermal Units per pound.

Example.—Initial temperature, 51·5°; final temperature, 63·7°; then $(63{\cdot}7 - 51{\cdot}5) \times 1{,}000 = 12{,}200$ B.Th.U. Evaporative power, $12{,}200 \div 970{\cdot}7 = 12{\cdot}5$ lb. from and at 212° F.

Gross and Net Calorific Value

The gross or higher calorific value of a fuel includes the heat evolved in the formation and condensation of steam from free and hygroscopic moisture and from the moisture formed by the combustion of hydrogen. In the calorimeter all the steam loses its latent heat and the moisture its sensible heat to the initial temperature of 60° F. Higher calorific value is given as "dry" and "as received." The latter is usually taken as the standard.

Net or lower calorific value takes into account the free and hygroscopic moisture and also the moisture due to the combustion of hydrogen by subtraction. A determination of net calorific value calls for an ultimate analysis of the fuel, although the hydrogen content is often based upon the percentage of volatile matter in the fuel, the latent heat being taken as 1,055 B.Th.U.s per pound.

CALORIFIC VALUE OF SOLID FUELS

Fuel	*Gross Calorific Value per lb.*		*Fuel*	*Gross Calorific Value per lb.*	
	Dry	*As received*		*Dry*	*As received*
	B.Th.U.	*B.Th.U.*		*B.Th.U.*	*B.Th.U.*
Straw	6,000	5,000	Coke	12,800	12,400
Oak bark	6,000	4,000	Coke breeze	11,000	10,000
Wood, dry	8,000	6,000	Midlands coal	12,600	11,500
Charcoal	13,000	12,000	Welsh smokeless	14,400	14,200
Peat, air dried	9,000	7,000	Anthracite	14,800	14,600

Analysis of Fuels.

The combustion engineer who is responsible for the efficiency of steam-raising plant must see that certain tests are made on the coal or other fuel which is being used. These tests are to determine:

(*a*) The calorific value of the fuel.
(*b*) The percentage of moisture.
(*c*) The amount of volatile matter.
(*d*) The percentage of fixed carbon.
(*e*) The percentage of ash.

He should also from time to time carry out tests to determine the amount of heat which is being carried away in the fuel gases, and should also see that periodic tests are made on the fuel gases to ensure that the fuel is being burnt as completely as possible. Perfect combustion of bituminous coal will give a flue gas comparison showing about 18 per cent. carbon dioxide.

Gas Testing

In order to ascertain the loss of heat in the waste gases of combustion with great accuracy, it is necessary to know the composition of the fuel, the weight

of waste gases per pound of fuel, and the initial and final temperatures. For most practical purposes the efficiency of combustion can be based upon the percentage of carbon dioxide in the gases and the final temperature. A simple formula for calculating heat losses is:

$$(t - 75) \times 2{\cdot}6 \times (CO_2 + 0{\cdot}75)$$

where t is the final temperature.

As an example, let the gas temperature be 500° F. and the CO_2 7 per cent. Then $(500 - 75) \times 2{\cdot}6 \times (7 + 0{\cdot}75) = 21$ per cent. loss.

If the percentage of CO_2 can be raised from 7 to 12 per cent. by careful attention to air supply and combustion conditions, the heat loss becomes:

$$(500 - 75) \times 2{\cdot}6 \times (12 + 0{\cdot}75) = 12{\cdot}8 \text{ per cent.,}$$

or a gain of $21 - 12{\cdot}8 = 8{\cdot}2$ per cent.

In practice, with the small plant, the percentage is seldom greater than 12 per cent. and should not drop much below 10 per cent. The percentage of carbon monoxide can be ascertained by means of the Orsat apparatus, but it is usually neglected as unlikely to be present with normal CO_2 readings.

The Orsat Gas-testing Apparatus

The Orsat apparatus is simple in use and with constant practice gives the highest degree of accuracy. The price is so low that it brings it within reach of the small fuel consumer, and while it is not so convenient as the fully automatic indicator or recorder, it can be strongly recommended for use in plants where the much more expensive automatic apparatus would be difficult to justify.

The complete apparatus can be obtained with two, three, or four absorption vessels, the type most frequently used being illustrated in Fig. 2. This is known as the "Orsat-Fischer," and consists of a case with a sliding back and front and containing a stopcock tube with four stopcocks; a graduated burette with jacket; absorption U-tubes; rubber bellows; and absorption U-tubes with copper spirals for cuprous chloride solution.

The measuring tube (A) contains from zero mark at the bottom to the upper capillary end exactly 100 c.c., but its graduation in $\frac{1}{5}$ c.c. only extends to 40 c.c. and ceases where the tube is enlarged. In order to protect the gas which is contained in this burette from the influences of the changes in temperature externally, the tube is surrounded by a water jacket, closed at the top and bottom by indiarubber stoppers, and provided with a white background of opaque glass upon which the black divisions of the burette are plainly visible. The bottom of the burette is connected by a rubber tube with a level bottle (B) filled two-thirds with water; the top end is connected to a glass capillary (A) bent at right angles and ending in a three-way cock. This tube is protected against breaking by a wooden frame, and carries at a right angle three taps h', h'', h''', each provided with a capillary tube and connected by indiarubber joints with the three U-shaped absorption vessels C', C'', C''', filled with bundles of glass tubes. The first of these is filled with a solution of caustic potash, the second with an alkaline solution of parogallol, the third with a concentrated

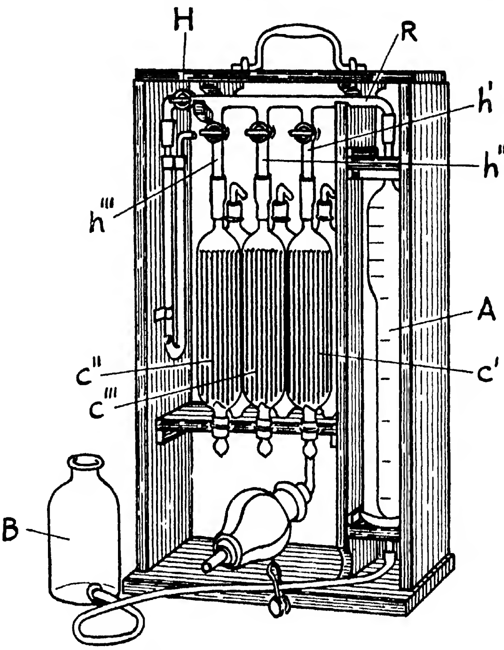


FIG. 2.—THE "ORSAT-FISCHER" GAS-TESTING APPARATUS

solution of cuprous chloride in hydrochloric acid. In order to keep this solution in an unchanged state, it is left in constant contact with copper spirals, introduced into the glass tubes with which the vessel C'' is filled.

The absorption vessels are filled with water slightly more than half-way up, and this is drawn up to the mark made in the capillary neck by opening the connecting tap and running off the water contained in the burette (*A*), for which purpose the levelling bottle (*B*) must be lowered. In order to protect the absorbing solutions against action of the air, the outer ends of the vessels are closed by small balls of thin rubber.

Manipulation

To obtain a sample of gas a length of iron or copper tube is inserted into the flue of the boiler with the inlet end as near as possible in the centre of the gas flow. This tube is then connected to the testing apparatus and a sample of the gas is drawn into the sampling tube either by means of a rubber bellows or a small hand-pump. The procedure is then as follows. Raise the level bottle (*B*), open the tap, and allow the burette (*A*) to fill with water up to the capillary part. Now aspirate the gas by lowering the level bottle and turning the tap through 90°. Run off the water a little below the zero mark and close the tap, compress the gas by raising the level bottle till the water rises above zero, squeeze the connecting rubber tube close to the joint, and then, after lowering the level bottle, allow the excess of water to run out to zero by cautiously loosening the rubber tube. Last of all, the tap is opened for an instant in order to produce a pressure equal to the atmosphere, whereupon exactly 100 c.c. of gas will be confined within the burette.

Now the absorption begins, first that of the carbon dioxide by conveying

of waste gases per pound of fuel, and the initial and final temperatures. For most practical purposes the efficiency of combustion can be based upon the percentage of carbon dioxide in the gases and the final temperature. A simple formula for calculating heat losses is:

$$(t - 75) \times 2{\cdot}6 \times (CO_2 + 0{\cdot}75)$$

where t is the final temperature.

As an example, let the gas temperature be 500° F. and the CO_2 7 per cent. Then $(500 - 75) \times 2{\cdot}6 \times (7 + 0{\cdot}75) = 21$ per cent. loss.

If the percentage of CO_2 can be raised from 7 to 12 per cent. by careful attention to air supply and combustion conditions, the heat loss becomes:

$$(500 - 75) \times 2{\cdot}6 \times (12 + 0{\cdot}75) = 12{\cdot}8 \text{ per cent.,}$$

or a gain of $21 - 12{\cdot}8 = 8{\cdot}2$ per cent.

In practice, with the small plant, the percentage is seldom greater than 12 per cent. and should not drop much below 10 per cent. The percentage of carbon monoxide can be ascertained by means of the Orsat apparatus, but it is usually neglected as unlikely to be present with normal CO_2 readings.

The Orsat Gas-testing Apparatus

The Orsat apparatus is simple in use and with constant practice gives the highest degree of accuracy. The price is so low that it brings it within reach of the small fuel consumer, and while it is not so convenient as the fully automatic indicator or recorder, it can be strongly recommended for use in plants where the much more expensive automatic apparatus would be difficult to justify.

The complete apparatus can be obtained with two, three, or four absorption vessels, the type most frequently used being illustrated in Fig. 2. This is known as the "Orsat-Fischer," and consists of a case with a sliding back and front and containing a stopcock tube with four stopcocks; a graduated burette with jacket; absorption U-tubes; rubber bellows; and absorption U-tubes with copper spirals for cuprous chloride solution.

The measuring tube (A) contains from zero mark at the bottom to the upper capillary end exactly 100 c.c., but its graduation in $\frac{1}{5}$ c.c. only extends to 40 c.c. and ceases where the tube is enlarged. In order to protect the gas which is contained in this burette from the influences of the changes in temperature externally, the tube is surrounded by a water jacket, closed at the top and bottom by indiarubber stoppers, and provided with a white background of opaque glass upon which the black divisions of the burette are plainly visible. The bottom of the burette is connected by a rubber tube with a level bottle (B) filled two-thirds with water; the top end is connected to a glass capillary (A) bent at right angles and ending in a three-way cock. This tube is protected against breaking by a wooden frame, and carries at a right angle three taps h', h'', h''', each provided with a capillary tube and connected by indiarubber joints with the three U-shaped absorption vessels C', C'', C''', filled with bundles of glass tubes. The first of these is filled with a solution of caustic potash, the second with an alkaline solution of parogallol, the third with a concentrated

solution of cuprous chloride in hydrochloric acid. In order to keep this solution in an unchanged state, it is left in constant contact with copper spirals, introduced into the glass tubes with which the vessel C'' is filled.

The absorption vessels are filled with water slightly more than half-way up, and this is drawn up to the mark made in the capillary neck by opening the connecting tap and running off the water contained in the burette (A), for which purpose the levelling bottle (B) must be lowered. In order to protect the absorbing solutions against action of the air, the outer ends of the vessels are closed by small balls of thin rubber.

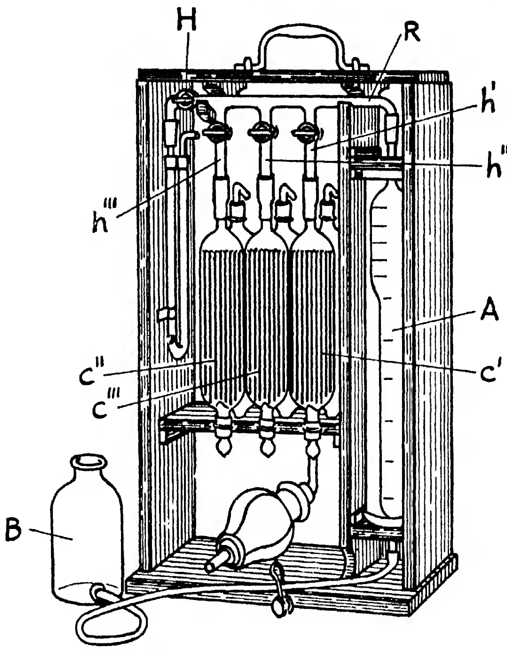


FIG. 2.—THE "ORSAT-FISCHER" GAS-TESTING APPARATUS

Manipulation

To obtain a sample of gas a length of iron or copper tube is inserted into the flue of the boiler with the inlet end as near as possible in the centre of the gas flow. This tube is then connected to the testing apparatus and a sample of the gas is drawn into the sampling tube either by means of a rubber bellows or a small hand-pump. The procedure is then as follows. Raise the level bottle (B), open the tap, and allow the burette (A) to fill with water up to the capillary part. Now aspirate the gas by lowering the level bottle and turning the tap through 90°. Run off the water a little below the zero mark and close the tap, compress the gas by raising the level bottle till the water rises above zero, squeeze the connecting rubber tube close to the joint, and then, after lowering the level bottle, allow the excess of water to run out to zero by cautiously loosening the rubber tube. Last of all, the tap is opened for an instant in order to produce a pressure equal to the atmosphere, whereupon exactly 100 c.c. of gas will be confined within the burette.

Now the absorption begins, first that of the carbon dioxide by conveying

THE TREATMENT OF WATER FOR BOILER PURPOSES

MODERN boiler practice will not tolerate hard water; the scale-forming elements must be removed by some form of water treatment or conditioning. It would be impossible to run a modern power-station steam generator for any length of time with untreated water without fear of overheating and collapse. Every engineer knows that boiler stoppages for scale removal are expensive, but with hard water they cannot be avoided. The type of treatment employed depends to a great extent upon the kind of impurity found in the water and the degree of hardness.

It is well known that the quality of water varies in different parts of the country, and it should be obvious that no substance or combination of substances could effectively counteract all the impurities to be found in water without previously knowing what they are or what proportion they represent. It is therefore an essential preliminary, in all cases where a water-treatment system is considered desirable, that an analysis of the water be obtained. When a public supply is under consideration, it is usually possible to obtain analyses from the local Medical Officer of Health or from the water company; failing this, a sample can be sent to one of the public fuel- and water-testing laboratories, who will supply a detailed report for a small fee.

Water Impurities

Pure water is seldom found in Nature, and even rain-water may be contaminated in its passage through the atmosphere. Rain, when it reaches the earth, invariably comes into contact with some form of mineral, gas, vegetable matter, or other impurities from which compounds may be formed and taken into solution.

For industrial purposes the most prevalent and important impurities found in water are the carbonates of lime and magnesium and the more stable sulphates of lime and magnesium. Water may also contain chlorates and nitrates, as well as carbon dioxide, oxygen, grease, and insoluble matter.

Definition of Hardness

Water is said to be "hard" when it is difficult to produce a lather with soap, and "soft" when a lather can be easily and quickly obtained. What is termed "temporary hardness" is due to the presence of carbonates of lime and mag-

nesium held in solution by carbonic acid and which precipitates at a temperature of 212° F. or less.

"Permanent hardness" is understood to be that part of the hardness which cannot be removed by boiling at atmospheric pressure, and is mainly due to the presence of sulphates of lime and magnesium and less commonly to chlorides and nitrates. "Total hardness" is the sum of the temporary and permanent hardness.

Degrees of Hardness

Hardness is most frequently expressed in terms of degrees per gallon, 1° being equivalent to the hardness imparted to the water by the presence of 1 grain of calcium carbonate per gallon. Hardness can also be expressed in terms of parts of calcium carbonate per 100,000.

One gallon of water at 62° F. weighs 10 lb. or 70,000 grains, so that the relationship between the two systems can be given as: degrees of hardness $\times 10 \div 7 =$ parts of calcium carbonate per 100,000.

The following will give some idea of the weight of solids that would accumulate in a boiler if fed with untreated water containing 15° hardness and evaporating 5,000 gallons of water per hour for seventy-two hours. Then $5{,}000 \times 15 \times 72 \div 70{,}000 = 77 \cdot$ lb. This, of course, would not occur in practice, as, apart from any question of water treatment, a proportion of the feed would consist of condensate and periodical blow-down would remove a large part of the precipitate.

Testing for Hardness

For this purpose a standard soap solution is used of a strength which corresponds to N/50 acid, that is each cubic centimetre (c.c.) of soap solution is equivalent to 0·001 grm. of carbonate of lime. This means that when 70 c.c. of water are used for testing, each cubic centimetre of soap solution represents 1° of hardness per gallon.

Total Hardness

To determine the total hardness of a water sample, measure out 70 c.c. in a stoppered bottle and about 25 c.c. of soap solution in a burette. Add the soap solution 1 c.c. at a time, and shake the bottle vigorously after each cubic centimetre is added. Continue until a lather lasting five minutes has been obtained. The number of cubic centimetres of soap solution minus 1 will give the total hardness of the water in grains per gallon.

Temporary and Permanent Hardness

To determine the temporary hardness, 70 c.c. of water are measured into a flask and 4 drops of methyl-orange indicator added. Run in nitric acid (N/50) until the pale straw colour turns pink. With care this can be obtained to within

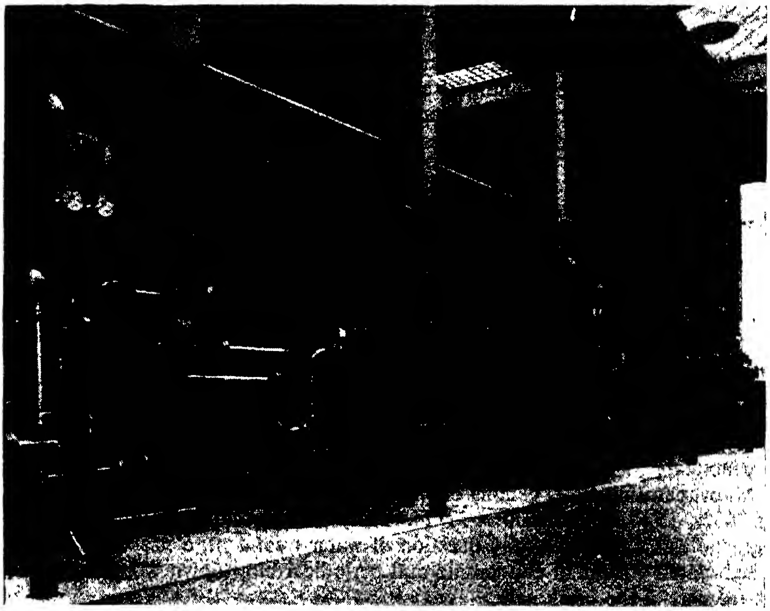


FIG. 1.—A PERMUTIT FULLY AUTOMATIC WATER-SOFTENING PLANT, WHICH TREATS 4,000,000 GALLONS DAILY (*Permutit Co., Ltd.*)

1 drop of acid. The number of cubic centimetres of acid will give the temporary hardness in degrees per gallon.

The permanent hardness is the total hardness minus the temporary hardness.

Alkalinity

The determination of the hardness of softened water is not by itself a proof of satisfactory treatment. A hardness of, say, 5° after treatment may mean that the water has been reduced to that figure and more reagent is required, or it can mean that an excess of reagent has been used and the water rehardened to 5°. To ascertain whether water has been correctly treated it is necessary to test for alkalinity.

To test for alkalinity, measure out 70 c.c. of the treated water and add 4 drops of purple indicator. Run in standard acid N/50 until the purple colour changes to blue, and note number of cubic centimetres required. Continue the addition until the blue turns through green to a permanent yellow, and note total number of cubic centimetres used. The number of cubic centimetres of acid required to obtain the blue colour gives the caustic alkalinity plus half the alkalinity due to carbonate of soda, whereas the final figures give the total alkalinity due to all the caustic plus all the carbonate; if, therefore, the final

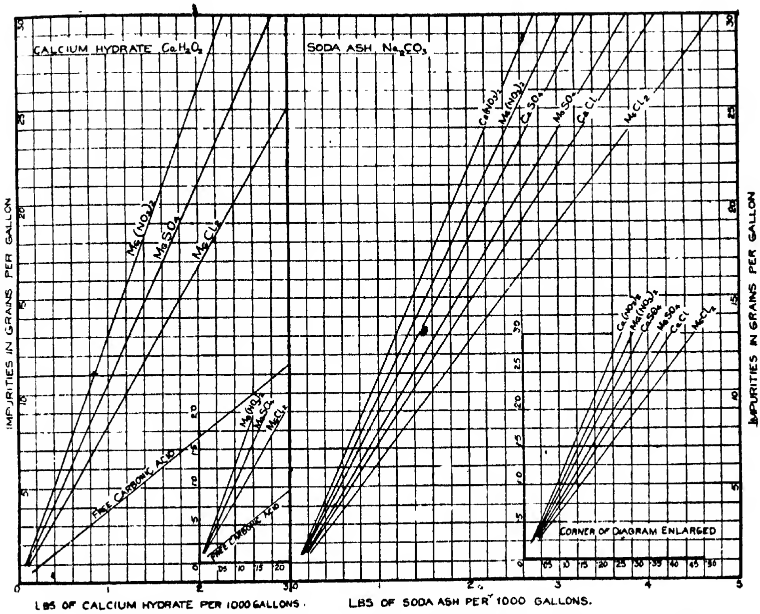


FIG. 2.—DIAGRAM SHOWING WEIGHT OF REAGENT PER 1,000 GALLONS OF FEED WATER TO BE SOFTENED

figure is less than twice the first figure, there is both caustic and carbonate present in the water and the water has been correctly treated.

Water Treatment

It is safe to say that all water-tube and smoke-tube boilers should be fed with water containing not more than 2°–4° of total hardness. In effect this means that the great majority of such plants should be equipped with a water-softening system.

In the case of Lancashire, Cornish, and vertical boilers, it is not always a simple matter to decide whether a satisfactory external water-softening plant could be justified. While the deleterious and wasteful effect of scale is frequently exaggerated, the fact remains that better heat conductivity will be obtained with clean heating surfaces than would be the case with surfaces coated with scale, and the thermal efficiency must therefore be higher. A great deal depends upon the proportion of make-up feed. In some factories a considerable amount of steam is used in process work, and the make-up feed represents a large proportion of the total feed water, while in others, where exhaust-steam feed heaters are installed, the amount of make-up feed may be small.

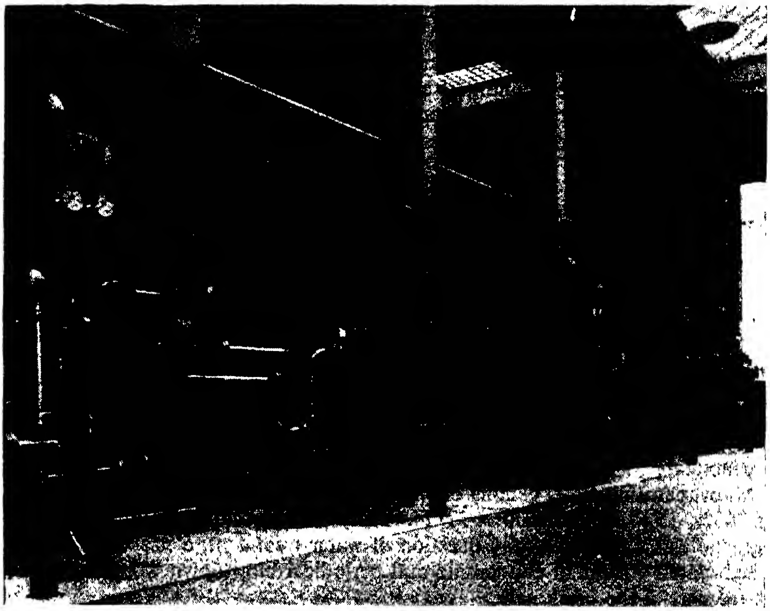


FIG. 1.—A PERMUTIT FULLY AUTOMATIC WATER-SOFTENING PLANT, WHICH TREATS 4,000,000 GALLONS DAILY (*Permutit Co., Ltd.*)

1 drop of acid. The number of cubic centimetres of acid will give the temporary hardness in degrees per gallon.

The permanent hardness is the total hardness minus the temporary hardness.

Alkalinity

The determination of the hardness of softened water is not by itself a proof of satisfactory treatment. A hardness of, say, 5° after treatment may mean that the water has been reduced to that figure and more reagent is required, or it can mean that an excess of reagent has been used and the water rehardened to 5°. To ascertain whether water has been correctly treated it is necessary to test for alkalinity.

To test for alkalinity, measure out 70 c.c. of the treated water and add 4 drops of purple indicator. Run in standard acid N/50 until the purple colour changes to blue, and note number of cubic centimetres required. Continue the addition until the blue turns through green to a permanent yellow, and note total number of cubic centimetres used. The number of cubic centimetres of acid required to obtain the blue colour gives the caustic alkalinity plus half the alkalinity due to carbonate of soda, whereas the final figures give the total alkalinity due to all the caustic plus all the carbonate; if, therefore, the final

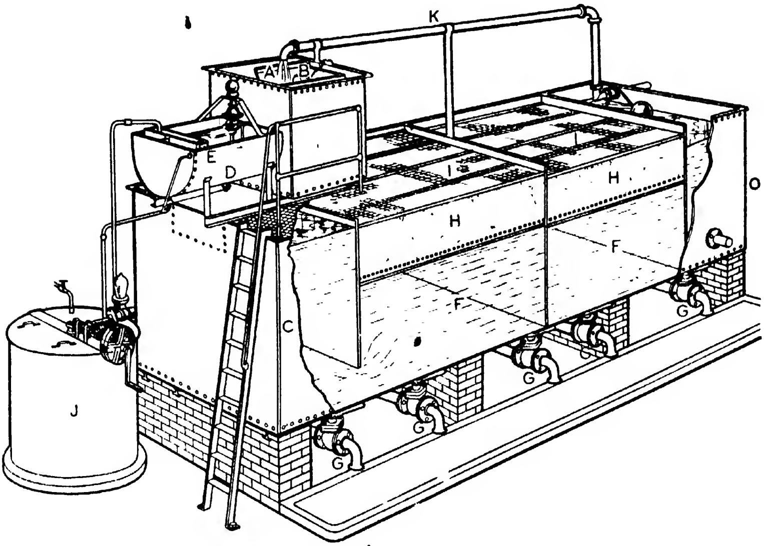


FIG. 3.—AN AUTOMATIC WATER SOFTENER WITH REAGENT TANK

phosphate, and trisodium phosphate; each giving different boiler alkalinities under otherwise similar conditions. Sodium metaphosphate has certain advantages over the others, as it can be passed through most feed lines without the risk of premature deposition of calcium phosphate, and gives lower boiler alkalinities.

Since in phosphate treatment all the insoluble solids form a sludge, it is clearly undesirable to use this process for feed waters with a high hardness unless the blowdown is heavy enough to prevent sludge accumulations.

Sodium metaphosphate can usually be added to the feed water on the suction side of the feed pump. The other two phosphates should be pumped direct into the boiler, unless the feed water consists of condensate and evaporated make-up.

COLLOIDAL TREATMENT.—Colloids are minute insoluble particles obtained from substances such as starch, linseed, oak, bark, and glue. When introduced into a boiler, the scale-forming salts become enveloped and absorbed in the colloid and precipitate as a non-adherent sludge. One particular system consists of permeating linseed with steam under pressure and allowing the resulting emulsion to flow into the feed tank. About 1 lb. of linseed is used for approximately 1,200 gallons of water. When first introduced into a boiler this emulsion undoubtedly tends to lift old scale and disintegrate it. The formation of new scale is also prevented, provided periodical blowdown is practised. The control does not appear to be exact, and little appears to be known as to the connection

between the amount of colloid and the proportion and nature of the impurities. As with other internal treatments, the amount of solids passing into the boiler is actually increased.

WATER-SOFTENING SYSTEMS

The Lime-soda Process

A complete water softener with chemical mixing tank is illustrated in Fig. 3. This is manufactured by Messrs. Babcock & Wilcox, Ltd., and is entirely automatic. The water to be softened enters pipe *K* and passes into the automatic measuring apparatus, by which the chemical reagents are added in proportion to the quantity of water to be treated. The water alternately fills one of the two compartments of the receiver *A*, which, when full, tips over, emptying its contents into the softening compartment *C*, and at the same time bringing the other compartment of the receiver under delivery pipe *K*.

The container *D*, fixed on the side of the receiver, is filled with the chemical solutions. With each oscillation of the receiver *A*, a double-beat valve in the bottom of the container *D* is opened, permitting a measured quantity of the solution to pass into compartment *C*, meeting the stream of hard water and becoming mixed with it.

The delivery of the chemical valve can be adjusted to suit the particular water being treated, and a definite amount of solution is added at each oscillation of the receiver, each oscillation being caused by a definite quantity of water, irrespective of the speed at which the water enters.

The reaction takes place in compartment *C*, where, if exhaust steam is available, a heater is usually fixed. The water flowing in the direction indicated passes upwards through filters of wood wool, which is packed tightly between two rows of wooden bars, into the storage tank *O*. The purified and softened water is drawn off through the outlet at the bottom of the tank.

The heavier precipitate is from time to time sludged off through the cocks *G* at the bottom of the settling tank, while, by removing the top bars of the wood-wool filters, the filter medium can be taken out, cleansed, and replaced as required.

Method of Determining the Amount of Reagent for Softening Feed Water

The weight of reagent in pounds required for the treatment per 1,000 gallons of water can be obtained from the diagram (Fig. 2). Each impurity generally encountered is represented by a diagonal line. The grains per gallon are found on the vertical scales, and the theoretical pounds of reagent on the horizontal scales.

The following example illustrates the use of the diagrams:

Analysis of Water

Sulphate of lime	27·05 grains per gallon.
Carbonate of lime	9·10 grains per gallon.
Sulphate of magnesia	6·81 grains per gallon.
Nitrate of magnesia	6·43 grains per gallon.
Chloride of magnesia	4·03 grains per gallon.
Free carbonic acid	6·31 grains per gallon.

The reagents to be used are as follows:

To treat carbonic acid	1·64 lb. calcium hydrate.
To treat sulphate of magnesia	0·65 lb. calcium hydrate.
To treat nitrate of magnesia	0·50 lb. calcium hydrate.
To treat chloride of magnesia	0·48 lb. calcium hydrate.
Total	3·27 lb. calcium hydrate.
To treat sulphate of lime	3·00 lb. soda ash.
To treat sulphate of magnesia	0·85 lb. soda ash.
To treat nitrate of magnesia	0·65 lb. soda ash.
To treat chloride of magnesia	0·64 lb. soda ash.
Total	5·14 lb. soda ash.

By treating the water with 10 per cent. excess reagent over that given—that is, $3·27 + 0·33 = 3·6$ lb. of calcium hydrate, and $5·14 + 0·51 = 5·65$ lb. of soda ash—the hardness of the water would be reduced to approximately 2° with a soda alkalinity of 2°.

The Hot Lime-soda Process

The hot lime-soda process is similar to the cold process in regard to the reagents used, but differs in that the reactions are carried out as near boiling-point as possible. The plant is more compact, with smaller settling tanks, and the speed of the chemical reactions increases rapidly as the temperature rises. An excess of reagents is required in both the hot and cold processes, but with the former it is possible to reduce the hardness of the water to about 2° per gallon with a smaller excess of lime and soda.

When exhaust or waste steam is available for heating purposes and the water to be treated contains a large proportion of temporary hardness, the hot process is often the most economical system.

Lime-Zeolite Process

Preliminary treatment by lime, followed by base-exchange softening, may be adopted with waters high in bicarbonate hardness. This combination system is mainly used when water is required for both boiler feed and cooling purposes in engine jackets. All the water is first treated with lime, which removes the bulk of the bicarbonate hardness and reduces the total solids. The permanent hardness is not reduced. The addition of a coagulant, such as sulphate of alumina, aids in settling the precipitates. The steam of pretreated water is then divided. One portion is completely softened by passing it through the zeolite softener, and is used for boiler feed, and the other portion is then neutralised with sulphuric acid until only about 1 grain of temporary hardness per gallon remains, and this is used for cooling purposes.

Base-exchange Process

A fully automatic base-exchange softener is illustrated in Fig. 1. This represents a "Permutit" apparatus, which is completely self-regenerating, and

between the amount of colloid and the proportion and nature of the impurities. As with other internal treatments, the amount of solids passing into the boiler is actually increased.

WATER-SOFTENING SYSTEMS

The Lime-soda Process

A complete water softener with chemical mixing tank is illustrated in Fig. 3. This is manufactured by Messrs. Babcock & Wilcox, Ltd., and is entirely automatic. The water to be softened enters pipe *K* and passes into the automatic measuring apparatus, by which the chemical reagents are added in proportion to the quantity of water to be treated. The water alternately fills one of the two compartments of the receiver *A*, which, when full, tips over, emptying its contents into the softening compartment *C*, and at the same time bringing the other compartment of the receiver under delivery pipe *K*.

The container *D*, fixed on the side of the receiver, is filled with the chemical solutions. With each oscillation of the receiver *A*, a double-beat valve in the bottom of the container *D* is opened, permitting a measured quantity of the solution to pass into compartment *C*, meeting the stream of hard water and becoming mixed with it.

The delivery of the chemical valve can be adjusted to suit the particular water being treated, and a definite amount of solution is added at each oscillation of the receiver, each oscillation being caused by a definite quantity of water, irrespective of the speed at which the water enters.

The reaction takes place in compartment *C*, where, if exhaust steam is available, a heater is usually fixed. The water flowing in the direction indicated passes upwards through filters of wood wool, which is packed tightly between two rows of wooden bars, into the storage tank *O*. The purified and softened water is drawn off through the outlet at the bottom of the tank.

The heavier precipitate is from time to time sludged off through the cocks *G* at the bottom of the settling tank, while, by removing the top bars of the wood-wool filters, the filter medium can be taken out, cleansed, and replaced as required.

Method of Determining the Amount of Reagent for Softening Feed Water

The weight of reagent in pounds required for the treatment per 1,000 gallons of water can be obtained from the diagram (Fig. 2). Each impurity generally encountered is represented by a diagonal line. The grains per gallon are found on the vertical scales, and the theoretical pounds of reagent on the horizontal scales.

The following example illustrates the use of the diagrams:

Analysis of Water

Sulphate of lime	27·05 grains per gallon.
Carbonate of lime	9·10 grains per gallon.
Sulphate of magnesia	6·81 grains per gallon.
Nitrate of magnesia	6·43 grains per gallon.
Chloride of magnesia	4·03 grains per gallon.
Free carbonic acid	6·31 grains per gallon.

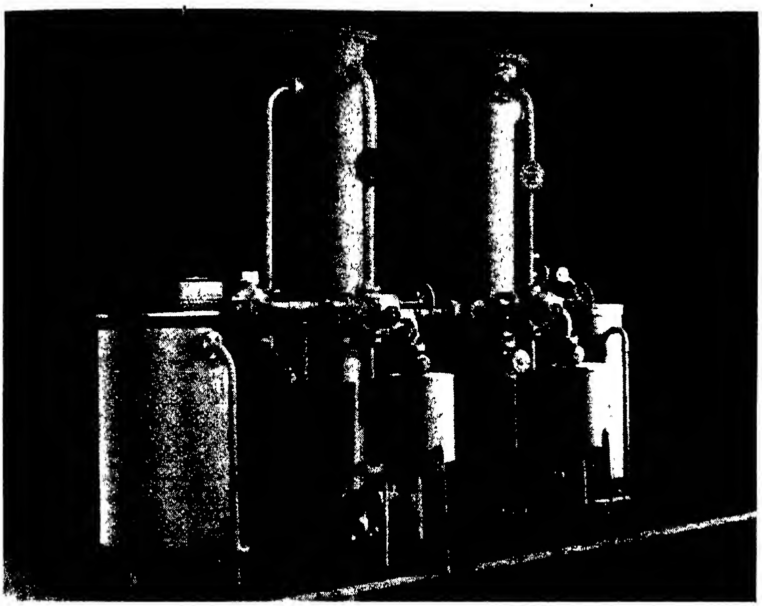


FIG. 5.—A TYPICAL "DEMINROLIT" PLANT FOR PRODUCING THE EQUIVALENT OF DISTILLED WATER BY ION EXCHANGE (*Permutit Co., Ltd.*)

The hydrogen-ion unit consists of a steel pressure tank with protective lining to contain the "Zeo-Karb," a tank to contain the regenerating material, and a closed degasifier to scrub out the CO_2 into which the bicarbonates have been transformed. The raw water flow is divided between this unit and a base-exchange water softener, the two being arranged to operate in parallel (Fig. 4). Simple automatic rate-of-flow controls maintain a constant ratio between the effluent flow rates so as to obtain the desired alkalinity in the final mixed effluent.

In another method only the "Zeo-Karb" unit is employed, and its effluent is neutralised with any suitable alkali, such as caustic soda, soda ash, or sodium phosphate, by means of any appropriate alkali feed in the degasifier influent or effluent line. This arrangement is usually employed for treating waters in which the chloride and sulphate content is relatively low, where the total volume of water treatment is small, or where the alkali is very cheap.

Another method, the "starvation treatment," is particularly suitable for dealing with waters high in bicarbonates and low in sulphates, chlorides, and nitrates. It is based on the principle that if "Zeo-Karb" material is regenerated with both acid and salt, it operates partly by hydrogen and partly by sodium

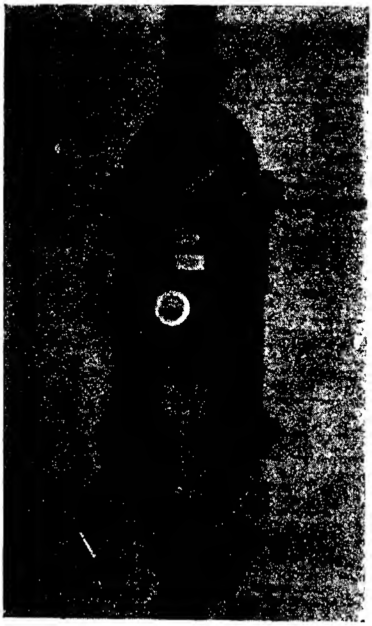


FIG. 6.—A 5,000-GALLONS-PER-HOUR "AQUASTAT" UNIT (*Aquastat, Ltd.*)

exchange, and the plant consists of a "Zeo-Karb" unit fitted with a dual regenerating system.

To provide demineralised water of a "distilled" quality suitable for high-pressure boiler feed, a "Deminrolit" plant may be used (Fig. 5). This process consists of two steps, the first of which is the use of a unit charged with "Zeo-Karb," as already described, which converts all the salts of sodium, calcium, and magnesium into corresponding acids. This acid effluent then flows through the second until charged with "Deacidite" material, which functions as an acid absorber. The mineral acids are there eliminated, and a water free from mineral salts and equivalent in purity to commercial distilled water is produced.

Electrical Water Conditioning

Electrical conditioning is simple and inexpensive, and is claimed to achieve complete freedom from corrosion troubles, although it does not, of course, produce water free from hardness. The process functions by means of light alternating current applied to electrodes housed in a container through which the water is directed (Fig. 6). The water is broken up into fine streams, which pass closely over the electrodes at a given rate. This electrical force modifies the electronic charge on the molecules of the hardness salts in solution, with the result that the normal process of crystallisation is interfered with. Instead of the usual hard scale formation, the precipitate comes down as a fine sludge.

Fig. 7 illustrates a section of the pipe unit of the "Aquastat" plant, which is inserted in the raw cold water line feeding the boiler. The water passing through the unit is subjected to electrical force supplied by a small control box, operated by battery or energised from A.C. mains. The unit consists of the outer casing and the inner electrode, which is concentric with and completely insulated from the casing. As will be seen, it has longitudinal slots milled throughout its length. It is to this electrode that the positive side of the electrical feed is applied, the negative going to the casing. The "Aquastat" automatically adjusts itself for variations in the flow of water.

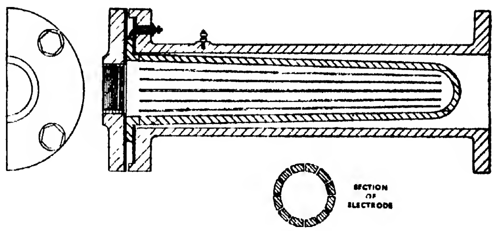


FIG. 7.—A SECTION OF THE PIPE UNIT OF THE "AQUA-STAT" PLANT (*Aquastat, Ltd.*)

De-oiling

Probably the most satisfactory method of dealing with emulsified oil in water is to use reagents that will entangle to coagulate the globules of oil in a precipitate that can afterwards be filtered out. The process usually adopted is to add small and fixed quantities of sulphate of alumina and carbonate of soda to the water, the result of the interaction of the substances being a flocculent precipitate of aluminium hydroxide which retains the oil and is removed by filtration, and a small quantity of sulphate of soda which, together with a small excess of carbonate of soda, remains in solution in the de-oiled water.

Mechanical oil separators will remove about 95 per cent. of the oil from exhaust steam, but oil globules in an emulsified state are microscopic in size and cannot be filtered out.

Caustic Embrittlement

Caustic embrittlement is indicated when minute cracks appear on the surface of boiler plates adjacent joints, seams, and rivet holes, below water-level. Plates attacked in this manner become brittle and the tensile strength is reduced.

The primary cause is said to be excessive stress during construction, and the secondary cause, the action of highly concentrated alkaline water acting upon the crystalline structure of the stressed metal. While such processes as rolling, flanging, and riveting will set up stresses, the subsequent annealing treatment will relieve much of the stress. Lap-welded and other riveted joints, however, are always under stress to some extent, and unequal expansion and contraction take place in most boilers. It would appear that a properly constructed boiler would be free from embrittlement, whatever the condition of the feed, and, in fact, many thousands of boilers have given twenty years or more of useful life without the least sign of embrittlement. At the same time, however, several well-authenticated cases of caustic embrittlement have actually occurred.

Caustic soda is largely used in water-softening processes, and it is to this element in conjunction with stressed metal that intercrystalline cracking is attributed. Caustic soda has been used with all types of boilers for many years without ill-effects, and the most reasonable conclusion is that some other element is present in the water. Some waters, particularly from wells, contain a large number of impurities, and it is when such waters are treated with caustic soda that particular attention should be given to the possibility of embrittle-

ment. It is safe to say that any ordinary town drinking water reduced to less than about 3° of total hardness per gallon can be used in any type of industrial boiler working at 150 lb. pressure per square inch or less, without fear of caustic embrittlement.

To prevent caustic embrittlement, if it is suspected, sufficient sodium sulphate should be used in the water-treatment system to obtain a ratio of one of sulphate to one of carbonate, at the same time keeping the alkalinity of the boiler water below 50 grs. of caustic soda per gallon by periodical use of the blowdown.

Corrosion

Corrosion in boilers is mainly due to the presence of dissolved oxygen in water coming into contact with the exposed surface of bare metal. It may also be caused by vegetable or mineral acids in the water, possibly accelerated by electrolysis.

The remedy is to expel the oxygen before it enters the boiler and to give the metal a protective bitumastic coating.

De-aerating.—Heating the water to as near boiling-point as possible will cause most of the oxygen and gases to be expelled, provided they can escape freely and time is given for this purpose. Slaked lime in the proportion of 1 lb. of lime to 500 gallons of water will assist in eliminating carbon dioxide.

Protective Coatings.—The success of this method depends upon an intimate association being obtained between the coating compound and perfectly clean metal. The metal should be well cleaned with a wire brush and graphite, or one of the special preservative paints should be well rubbed into the surface of the metal.

Concentration and Blow-down

Salts in solution and other impurities introduced into a boiler become concentrated as the water is evaporated. The safe limit of concentration depends upon the type of boiler, working pressure, rate of evaporation, and the composition of the feed water. The density of boiler water is ascertained by means of a specially designed hydrometer, and concentration of solids is reduced by periodical or continuous blow-down. Boilers of the Lancashire, Cornish, and vertical types, steaming at pressures up to 250 lb. per square inch, can be safely worked until the total dissolved solids have reached 750 parts per 100,000, while highly rated boilers, working at pressures of 500 lb. per square inch or more, should be limited to about 150 parts per 100,000.

E. P.

STEEL PIPES FOR STEAM, WATER, GAS, AND AIR

STEEL pipes may be divided into two types—welded and seamless. All pipes and fittings are made to conform to B.S. 534, 1934, and 1387, 1947, for water and gas pipes, and B.S. 806, 1942, in the case of steam pipes. Other sizes can, of course, be made to meet any special requirements. Various methods of manufacture are used: for welded pipes either the continuous-weld, roll lap-weld, or hydraulic lap-weld process is employed, and for seamless pipes either the push-bench, automatic, or Pilger process. Welded pipes for water and gas can also be made by the butt-welding method.

WATER, GAS, AND AIR PIPES

Protective Coatings

Gas and water pipes are protected internally and also externally, when laid out of doors. Internal protection for gas pipes usually consists of a coating of linseed oil, which is applied at the maker's works; but if the gas to be conveyed is undried, an anti-corrosive protection will be necessary as well.

For water mains the application of bitumen lining is strongly recommended for the protection of the internal surface against corrosion. This applies even where it would seem that under present conditions the water would have no detrimental effect on ordinary coated pipes.

STANDARD BITUMEN LINING.—This lining is applied centrifugally to the pipes after they have been completely descaled and coated.

THICK BITUMEN LINING.—This lining is of slightly different composition to that mentioned above, but it is applied to the pipes in a similar manner. It is now only used for exceptional conditions.

The types of external protection available for both water and gas pipes are as follows:

COATING.—This is a bituminous composition applied at the manufacturer's works, used for pipes laid above ground, or for pipes laid on ground where no corrosive action is likely to be experienced. It is usually considered that coating alone is not sufficient for buried mains.

SECURITY WRAPPING.—This is applied in three stages. The pipes are first coated and then covered with hot bitumen about $\frac{3}{64}$ in. thick. Finally, a layer of dried and chemically treated hessian cloth, impregnated with bitumen, is wrapped round the pipes. This covering will withstand corrosion, and is used on pipes of less than 4 in. bore.

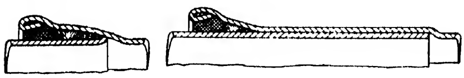


FIG. 1.—SPIGOT AND SOCKET JOINTS
(*Left*) Ordinary joint.
(*Right*) Long-sleeve inserted joint.

SHEATHING.—This is a tough bitumen-asbestos compound applied in varying thicknesses according to the size of pipe. It is suitable for pipes of 4-in. bore and larger.

For external protection of the pipe joints the following method may be used. Removable moulds are placed round the joint, and these should extend over the protective covering on each side of the joint so that a continuous covering is ensured. A molten bituminous composition is then poured into the moulds and allowed to set. The moulds can then be removed.

Spigot and Socket Joints for Lead and Yarn

There are two principal forms of spigot and socket joint on unlined pipes, the ordinary sleeveless joint and one with a sleeve, shown in Fig. 1. These are generally used on pipes which are laid underground. It is advisable not to use the sleeveless joints for pipes laid above ground unless special attention is paid to the anchoring of the joint.

The sleeve type, in which both the sleeve and the spigot have a slight uniform taper, has the advantage that any exceptional stress to which the pipe may be subjected will not disturb the lead joint. When laid above ground it is usual, especially on the larger sizes, for the joints to be secured by bolts passed through lugs welded on the spigot and socket ends.

For water mains, the maximum working pressure for these joints is usually restricted to about 350 lb. per square inch for sizes up to 12 in., and to about 300 lb. per square inch for the larger sizes.

Spigot and socket joints for lead and yarn can be used for water, gas, and air mains, although the modern tendency is to use welded joints for gas and air mains wherever possible.

Welded Joints

These are specially suitable for gas and air mains, and can be used on pipes laid above ground or underground. There are three types of joint: the simple butt-welded joint, for which pipes are supplied with plain or bevelled ends

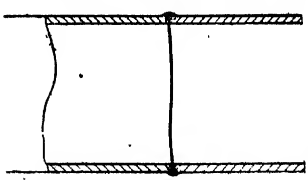


FIG. 2.—BUTT-WELDED JOINT

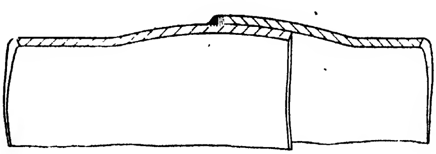


FIG. 3.—SPHERICAL-WELDED JOINT

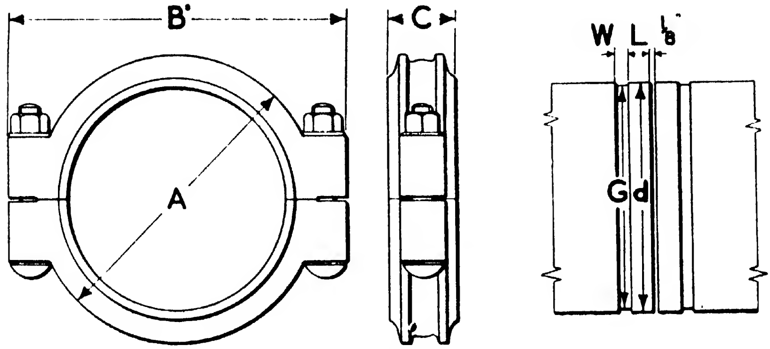


Fig. 4.—Victaulic joints for grooved tubes

according to thickness; the slightly tapered spigot and socket joint, and one with a spherical spigot and socket.

The tapered spigot and socket type of joint is suitable for long, straight stretches, or where heavy bending stresses occur on the pipes and joints. The spherical joint can be used where there is frequent deviation in the pipeline, since angles of 5° can be obtained between adjacent pipes with this form of joint during laying. An angle of 5° is equivalent to laying pipes in 20-ft. or 25-ft. lengths at 80 yards or 100 yards radius respectively, or 1 in. per foot out of line.

Victaulic Joints

Victaulic joints (Fig. 4) can be used on pipes laid underground or above ground, and are suitable for water, gas, and air mains. In various forms these joints are suitable for working pressures from below zero up to 1,500 lb. per square inch hydraulic.

The joints are easy to assemble, and can be put together on steel pipes at very low cost. When assembled they are automatically leak-tight, and any subsequent application of pressure or vacuum intensifies the seal. They allow a small

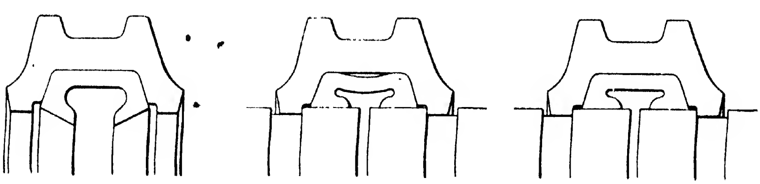


Fig. 5.—Sections through housing and rings
(*Left*) Before use. (*Centre*) Under Vacuum. (Right) Under pressure.

angular movement between adjacent pipes or a longitudinal movement of at least $\frac{1}{8}$ in. If Victaulic joints are used throughout the pipeline, the longitudinal movement provides for expansion and contraction on pipes laid above ground and for subsidence in underground mains.

The bolted housing is in two parts for sizes up to 14 in. and in four or more parts for the larger sizes. The toggle type of housing, in which no bolts are used, is specially suitable for temporary installations of mains requiring to be laid quickly. The parts of the housing are all hinged together, and when snapped round the joint are immediately locked by the toggle lever.

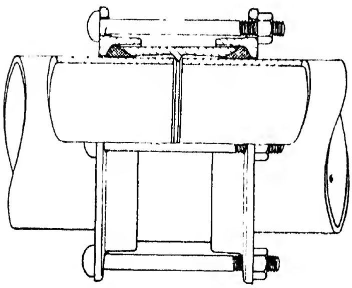


FIG. 6.—JOHNSON COUPLING

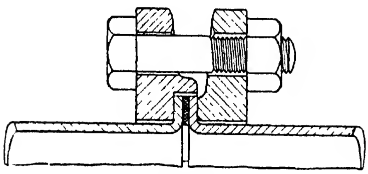


FIG. 7.—STEWARTS' LOOSE-FLANGE JOINT

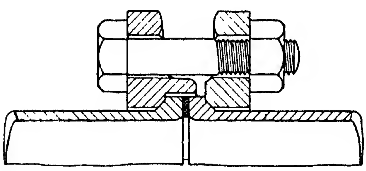


FIG. 8.—ALBION LOOSE-FLANGE JOINT

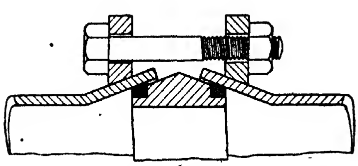


FIG. 9.—WILLIAMS' LOOSE-FLANGE JOINTS

Johnson Coupling

Standard Johnson couplings (Fig. 6) are available in sizes up to 30 in. for pressures up to 800-ft. head, but, if required, couplings of larger size or for higher pressures can be obtained. It is designed for pipes with plain ends, and is made so as to give considerable angular flexibility. It is suitable for water, gas, and air mains, and where frequent rearrangement of the mains may be required.

The joint consists of a centre sleeve and two end flanges, which hold two wedge-shaped packing rings in contact with the sleeve and pipe ends, enclosing and protecting them from damage. The sleeve normally has a central rib, as shown in Fig. 6, which acts as a locating stop. Sleeves without this rib are suitable for use as a closing connection, and also permit the removal of individual lengths without disturbance of the adjoining pipes. Special joint rings are supplied for use on gas mains.

The joint is sufficiently flexible to permit pipes of 20-ft. or 25-ft. length being laid 24 in. or 30 in. out of

line, or at 65 yards or 80 yards respectively. At the same time normal longitudinal movement of individual pipes can occur without leakage. In cases where the couplings are used throughout the main, provision is made for the distortion and lengthening of the line caused by stresses and subsidence.

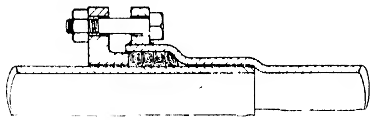


FIG. 10.—VULCAN JOINT

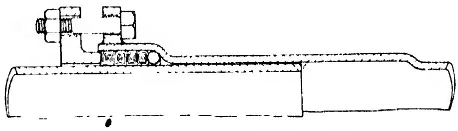


FIG. 11.—VULCAN EXPANSION JOINT

Loose-flange Joints

Three standard forms of loose-flange joints are shown in Figs. 7–9. These are known as the Stewarts', Albion, and Williams' joints: they are specially adapted for use in mines and other cases where pipelines are laid underground. The joints are suitable for water, gas, and air mains.

Joints in sizes up to 10 in. nominal bore are suitable for working pressures not exceeding 300 lb. per square inch for water and up to 150 lb. per square inch for air or gas.

For pipelines laid on the surface, the Vulcan joint illustrated in Fig. 10 may be used. The joint is based on the stuffing-box principle, the parallel sides of the packing space preventing the packing being forced out past the gland. If the pipes are separately anchored, each joint will take up the expansion and contraction of its own length of pipe. They are suitable for working water pressures up to about 500 lb. per square inch.

The Vulcan expansion joint (Fig. 11) is a modified version of the above. It can be used in conjunction with any rigid type of joint to provide for expansion and contraction.

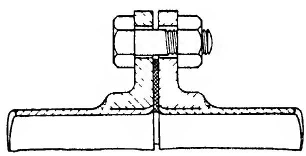


FIG. 12.—FLANGES WELDED ON AT BACK, WITH PIPE ENDS EXPANDED AT FACE

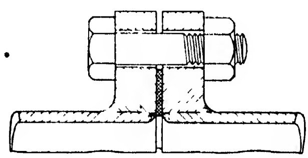


FIG. 13.—FLANGES WELDED ON BACK AND FRONT

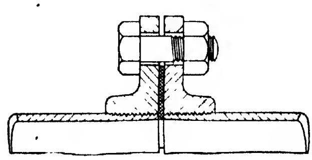


FIG. 14.—FLANGES SCREWED AND EXPANDED ON

Fixed-flange Joints

Welded-on flanges can be used on any thickness of pipe. They are cheaper

and more reliable than riveted-on flanges, which are now seldom used except as loose flanges for fixing at the site where welding apparatus is not available.

The screwed and expanded-on flange joint is usually adopted on pipe sizes of 6 in. and smaller, but can be used only on pipes thick enough to be screwed. Flanges are usually faced plain and drilled to the appropriate British Standard Specification.

Special Fittings

In addition to straight piping, there are many kinds of special fittings. These include bends, Ts, crosses, and branch pieces. Bends are manufactured with angles up to 90°, though the types most commonly used are those with angles of 11¼°, 22½°, 45°, and 90°.

There are also a number of fittings which are peculiar to the Victaulic joints mentioned on page 365. The more common of these are the following:

ADJUSTABLE BEND.—This consists of two special goose-necked setting pieces united by a standard Victaulic joint. The effect of the special shape of the setting pieces is that an alteration of the angle does not alter the plane in which the adjoining pipes lie. By swivelling the parts, any angle up to 45° can be obtained.

SHORT REDUCER.—The Victaulic short reducer is specially designed to

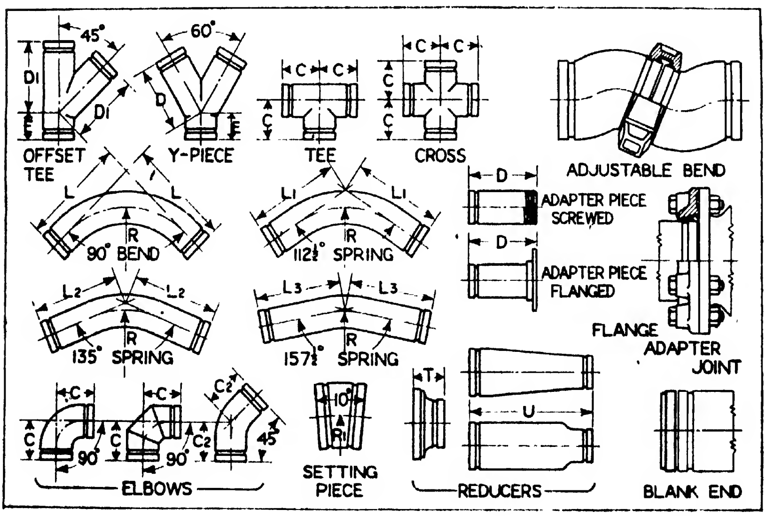


FIG. 15.—VICTAULIC FITTINGS FOR GROOVED TUBES

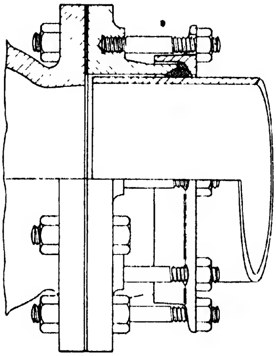


FIG. 16.—FLANGE ADAPTER FOR JOHNSON COUPLINGS

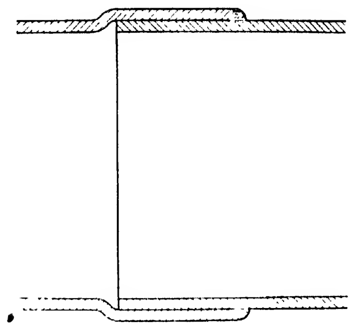


FIG. 17—STEWARTS' SLEEVE-WELDED JOINT

give a true nozzle flow, and consequently a negligible loss through friction while retaining the advantage of short length

FLANGE ADAPTER JOINT.—This joint is used in cases where a pipeline with Victualic joints joins another main having a different type of joint. It clips over the prepared end of the standard pipe, and is simply bolted on to the end flange of the other line. The joint is automatically sealed by a special ring.

HIGH-TEMPERATURE FLANGE JOINT.—The Victaulic standard joint should not be used on pipes where the temperature is likely to exceed 150° F. If temperatures above this point occur in a short portion of the line, e.g. between the compressor and the receiver on a compressed-air main, the high-temperature flange joint may be substituted for the ordinary joint. These joints clip on to the ends of the standard pipes without any further preparation, converting them into flanged pipes which can be bolted together in the usual way.

CLOSER JOINT.—This joint is designed to provide a liberal margin of error in the cutting of closing lengths. It should be noted that this is not an expansion joint, and that the pipes it connects are not coupled absolutely so that anchorage may be necessary to prevent blowing out.

Another fitting which deserves special mention is the flange adapter for Johnson couplings (Fig. 16). This is used in circumstances where a plain end pipeline with Johnson couplings has to be connected to a line with ends prepared for other forms of joints. Connectors with one end plain and the other end prepared to suit the particular joint can be used, but where the line to be connected is flanged, the flange adapter joint is preferable.

STEAM PIPES

Steel pipes for steam mains are divided into six classes. These are laid down in B.S. 806, 1942, as follows:

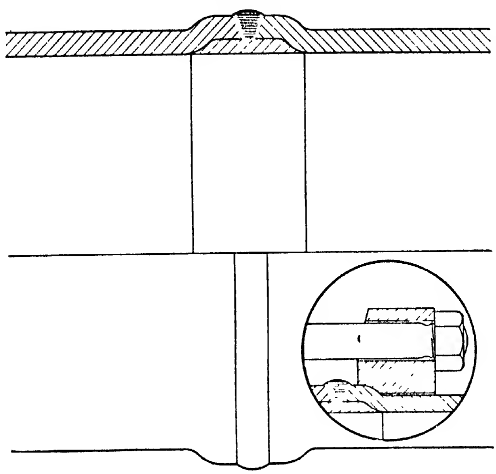


FIG. 18.—THE DAWSON JOINT

(*a*) Cold-drawn seamless steel pipes.

(*b*) Hot-finished seamless steel pipes.

(*c*) Hot-finished seamless steel pipes.

(*d*) Hydraulic (water gas) lap-welded steel pipes.

(*e*) Roll lap-welded steel pipes.

(*f*) Continuous-weld steel pipes.

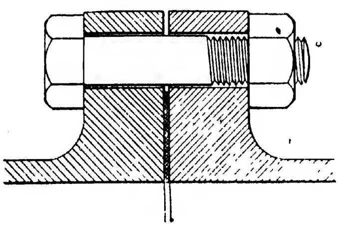
It should be noted that classes (*c*), (*e*) and (*f*) are suitable for pressures up to and including 300 lb. per square inch and temperatures up to and including 500° F.

All pipes are thoroughly tested before leaving the manufacturer's works. They are subjected to a tensile test, flattening test (for pipes up to and including 4½ in. outside diameter), cold bend test or expanding test.

Dawson Joint

This joint has been specially designed by Messrs. Stewarts and Lloyds, Ltd., to withstand high temperatures and pressures. It is an all-welded joint, in which the weld is built up between the pipe ends and carried down into the internal nipple, forming an extremely rigid combination (Fig. 18).

In erecting closer pipes with Dawson joints, it is usual to pull them up an amount equal to the cold allowance by expansion, and this provides sufficient space between adjacent tube ends to facilitate the insertion of the pipe over the sleeve. If desirable, further latitude may be obtained by leaving unwelded one or two joints farther along the main until the closer pipe is in place.



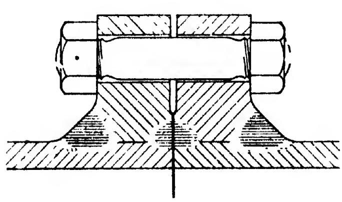


FIG. 19.—PLAIN-FACED FLANGES.

FIG. 20.—WELDED-ON FLANGE JOINT, TYPE "A"

Where pipes with Dawson joints connect to valves, it is necessary to provide on the valve end a projection similar to the pipe end. An internal thimble is provided as for ordinary pipe-to-pipe joints, and the connecting pipe, the valve end, and the thimble are welded together in the usual way.

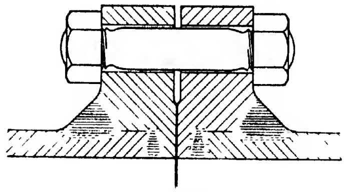


FIG. 21.—WELDED-ON FLANGE JOINT, TYPE "B"

Welded Joints

Fig. 17 shows the sleeve-welded joint, which is particularly suitable for lengthy steam mains. It has a slightly tapered spigot which is driven into the correspondingly tapered socket. This centres the pipes accurately and holds them in position during welding.

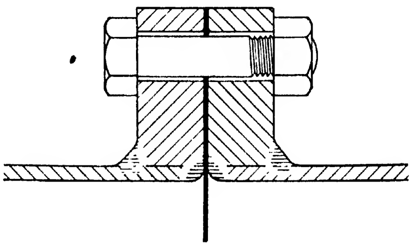


FIG. 22.—WELDED-ON FLANGE JOINT, TYPE "C"

The joint requires no maintenance, and has the advantage of enabling the pipe insulation to be continuous.

Flange Joints

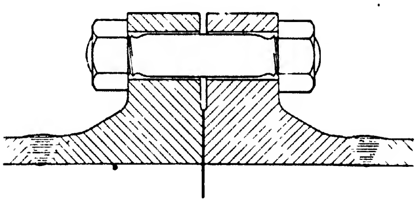


FIG. 23.—WELDED-ON FLANGE JOINT, TYPE "E"

Flanges may be fixed to the pipes by welding, screwing, expanding or riveting, or by a combination of two or more of these methods.

Plain, faced flanges (Fig. 19) are used for pressures up to 450 lb. per square inch, and for higher pressures the British Standards Institution have standardised flanges with facing strips. These strips, which should be not more than $\frac{1}{16}$ in. high, intensify the pressure of the flanges on the joint ring, and so help to keep the joint tight under high internal pressure.

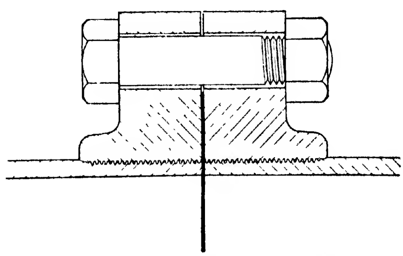


FIG. 24.—HEAVY SCREWED AND EXPANDED-ON FLANGE JOINT

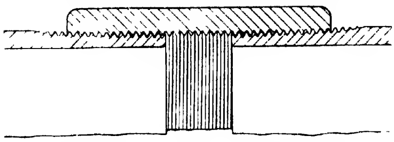


FIG. 25.—PARALLEL SCREWED SOCKET

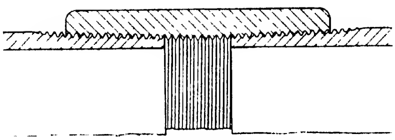


FIG. 26.—TAPER SCREWED SOCKET

Welded-on Flange Joints

There are three types of welded-on flange joints which may be used for high-pressure mains. These are shown in Figs. 20, 21, and 22, and cover all steam pressures and temperatures for which mild-steel pipes are suitable. The welding is done by the metallic arc process with covered electrodes.

For low pressure the type of joint shown in Fig. 23 is used. It is suitable for pressures up to and including 250 lb. per square inch at temperatures not exceeding 700° F. Flanges with hubs or bosses may be used in sizes not exceeding 12 in. nominal bore.

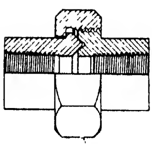


FIG. 27—ALL-STEEL SOCKET UNION

Screwed and Expanded-on Flanges

These joints have been used for pressures up to 350 lb. per square inch and temperatures up to 750° F. on pipes up to 12-in. bore. However, it is now largely superseded on pipe sizes over 4-in. bore by welded-on flanges.

When this type of joint is used, the precaution of welding at the back of the flange is sometimes recommended. This will give additional resistance against torsional stresses, but unless severe ones are likely, it will be sufficient to screw and expand on flanges without welding. The expanding is usually done at the manufacturer's works, but the smaller-size tubes may be screwed and expanded at the site.

Screwed and Socketed Joints

For small pipes the screwed and socketed joint is still popular. The tube ends are screwed with a taper thread to B.S. 21, 1938, while the sockets are either screwed parallel as in Fig. 25 or taper to correspond with the tube ends as in Fig. 26. The latter type is more expensive, but is suitable for more severe conditions. Long screws are not practicable for use with high-pressure steam, and socket unions must be employed instead.

Closing or Template Pipes

Template pipes should be chosen to be as few in number as possible, and for this reason bends are most suitable for selection as template pipes, since each bend provides adjustment on two stretches. In sending particulars of

template pipes to, the makers, there are two methods; these are: measurement and making templates.

If templates are used, the following precautions should be taken:

(1) The templates must be rigidly constructed.

(2) Flange discs must be stout and firmly fixed.

(3) After removing the flange bolts, the template should again be tried in place to check its accuracy before despatch.

(4) If wood is used, it must be thoroughly seasoned, as any warping would render the template useless.

(5) In making wooden templates, screws should be used, not driving nails.

Damage and distortion during transit will make the template useless. Whilst the actual damage is usually obvious, there is often no reason to suspect the accuracy of a template until the template pipe proves a misfit. It is therefore desirable that particulars of template pipes should be given by measurement.

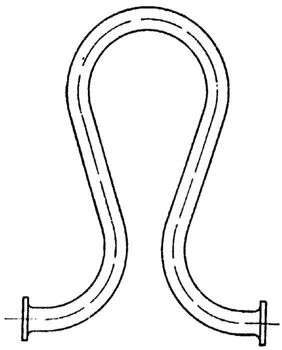
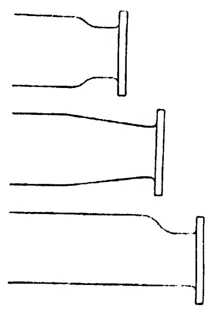


FIG. 28.—STANDARD EXPANSION BEND



FIGS. 29-31.—REDUCED ENDS OF PIPES

(*Top*) Short cress.
(*Centre*) Long taper.
(*Bottom*) Eccentric reduction.

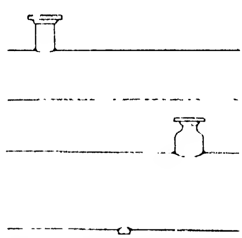


FIG. 32—(*Top*), WELDED-ON BRANCH PIECE

FIG. 33—PIECE WITH REDUCED ENDS

Expansion Bends and Joints

In order to provide for the expansion which occurs in a steam main, a sufficient number of ordinary bends should be fitted to enable the whole main, or each portion into which it is divided by anchors, to take up all expansion and contraction by its own movement without the occurrence of excessive stresses. If a sufficient number of bends cannot be arranged, the expansion bend shown in Fig. 28 may be used. Special forms of these bends can be obtained for use in positions other than in straight runs, and for where space is limited.

Sliding expansion joints of the stuffing-box type are used in similar stretches where space does not permit the use of expansion bends, but this type should not be employed for high-pressure steam mains. In certain cases bellows

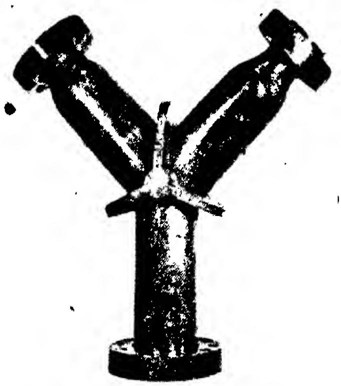


FIG. 34.—BRANCH PIECE WITH TRIFORM REINFORCEMENT

expansion joints can be employed for low-pressure work. When erecting an expansion joint, it will be necessary to spring it. Care should be taken to prevent any movement of the pipe at the anchors, as this would mean the loss of so much of the cold springing and consequently an increase in the stress imposed on the bend in the working condition if it still had to take up the same amount of expansion movement.

Pipe anchors in certain positions are essential for the satisfactory working of all types of expansion bends or expansion joints, and even where neither of these is employed, anchors may be necessary to control the movement of the pipe. Guides to maintain alignment are

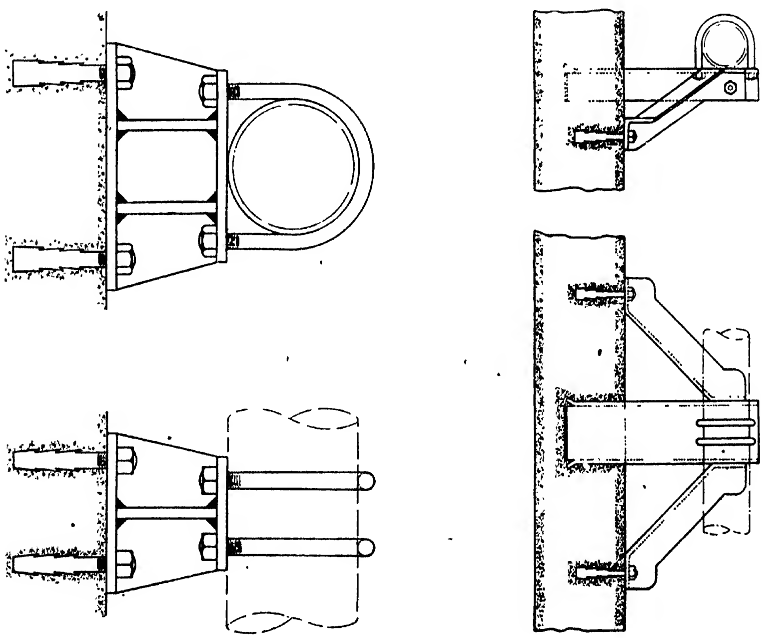


FIG. 35.—TYPES OF PIPE ANCHORS

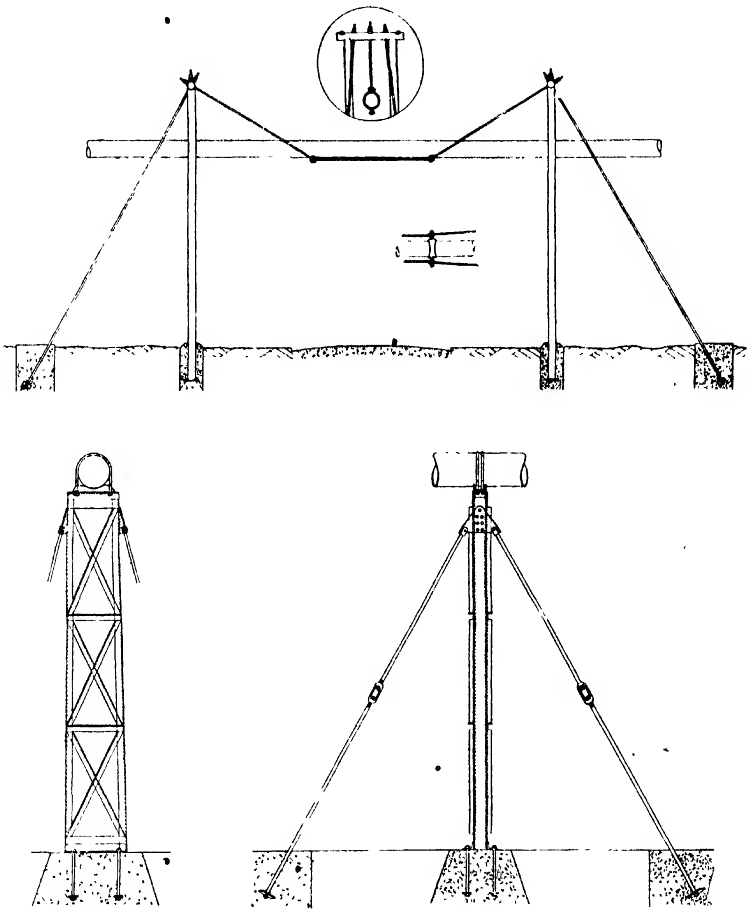


FIG. 36.—TYPES OF PIPE SUPPORTS

necessary for expansion joints, because any misalignment will probably cause the joint to jam.

Reduced Ends of Pipes

Three types of reduced ends of pipes are shown in Figs. 29-31. They are for use in circumstances where the bore of a main has to be reduced, e.g. where two sections of different diameters meet.

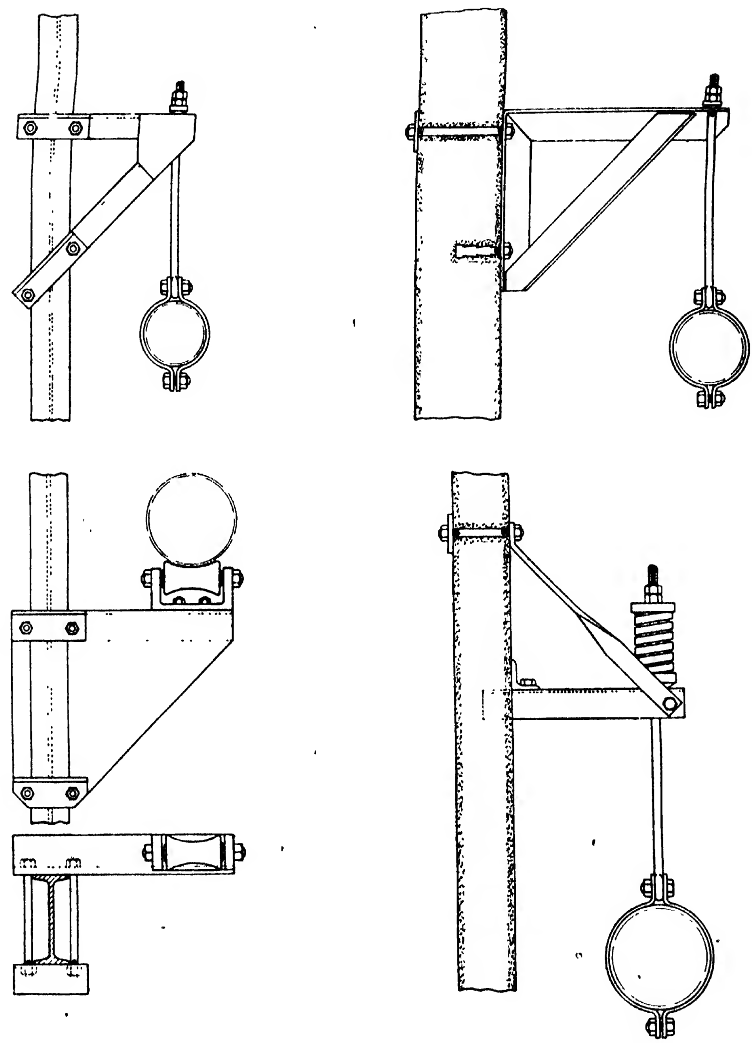


FIG. 37.—TYPES OF PIPE-BRACKET SUPPORTS

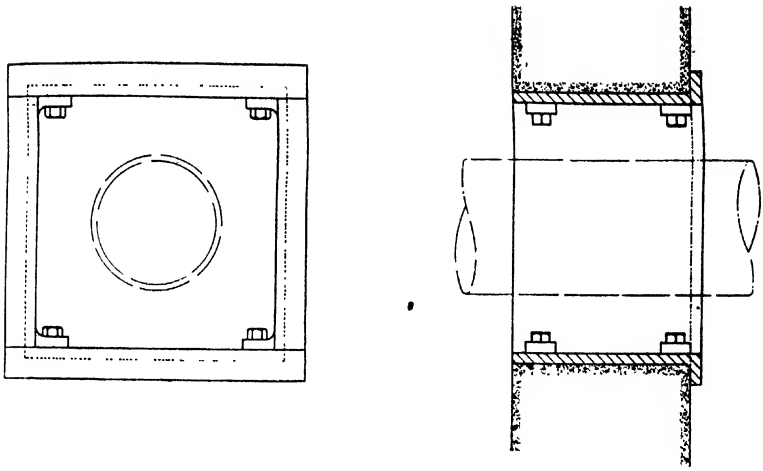


FIG. 38.—TYPE OF WALL BOX

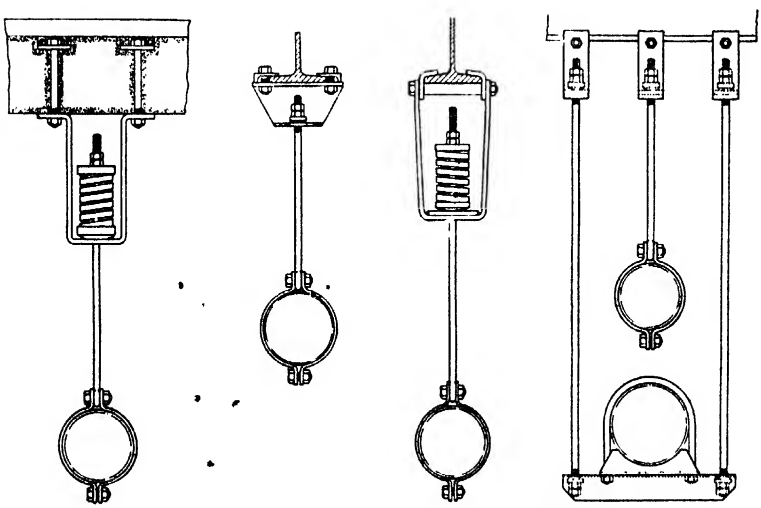


FIG. 39.—TYPES OF HANGER AND COLUMN SUPPORTS

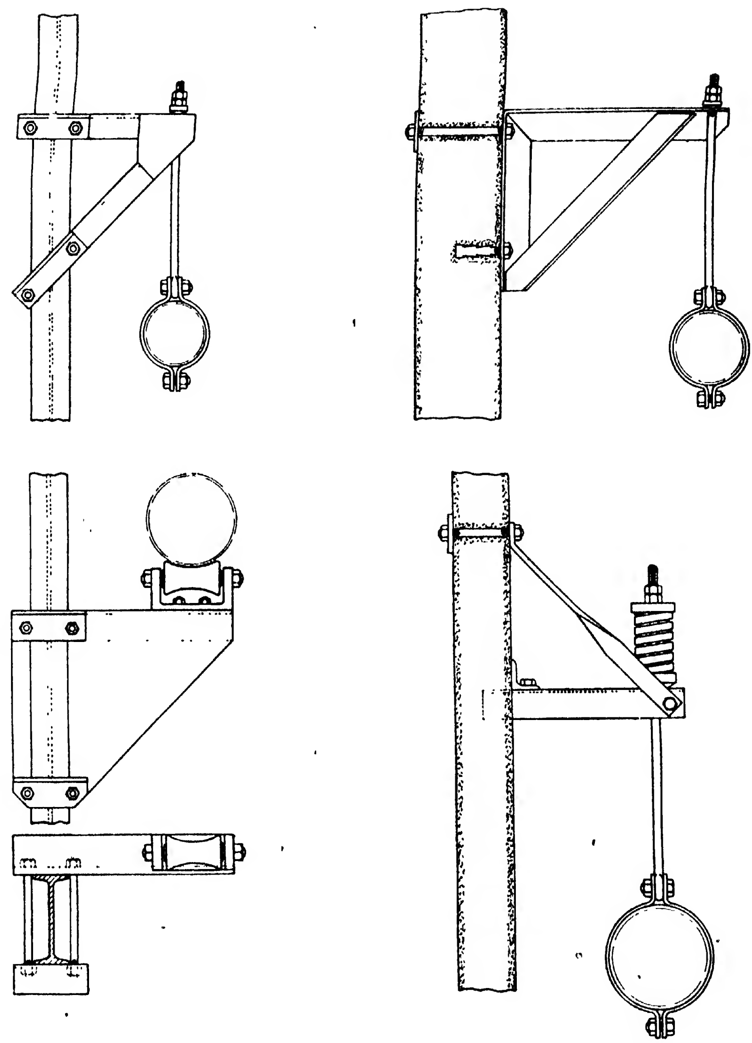


FIG. 37.—TYPES OF PIPE-BRACKET SUPPORTS

nal pressure, a reduced strength compared with that of a plain straight pipe of the same diameter and thickness. The amount of weakening depends largely on the size of the branch relative to the main pipe, and the angle between the branch and the main pipe. Where the branch and main pipes are the same diameter, the weakening must be taken into account, and in many other cases also it is advisable to give it consideration. Reinforcement may be obtained by one of two methods: the branch piece can be made thicker than the rest of the pipeline, or a metal collar or ring may be placed round, or near, the opening in the main pipe. It should be noted, however, that the first method is not always suitable for fabricated branches.

A special reinforcement shown in Fig. 34 has been designed by Messrs. Stewarts and Lloyds, Ltd. This reinforcement is designed primarily to strengthen the branch piece against internal pressure and it will also increase its strength against external forces. It is suitable for use on branch pieces fabricated by welding, since, although it can be applied to castings either as a welded-on reinforcement or as an integral part of the casting, these are usually more conveniently reinforced by the provision of additional thickness.

Drainage

When steam pipes are installed, attention should be paid to providing ample draining facilities for the condensed steam, because if the water is allowed to accumulate in the pipe, "water hammer" may result.

Drain connections should be installed in the mains at points where there are valves of smaller diameter than the main, expansion bends in vertical planes, at the foot of each rising bend, and in similar positions.

Supports for Steam Pipes

Various types of supports are shown in Figs. 35–40. Supports should permit free movement of the pipe in any direction, thus allowing for expansion and contraction. Spring supports should be used on all pipes which may be subject to vertical movement in order to avoid excessive stresses that might be set up in the pipes if rigid supports were used. Special care should be taken in the selection of springs, for if these are too weak they will allow vibration to take place, whilst if they are too rigid they will not relieve the stresses in the pipeline. Spacing of the supports will largely depend on the structural features of the building in which they are being installed, but, as a general rule, supports should be placed at intervals of not more than 10 ft. for a $\frac{3}{4}$-in. pipe, the maximum interval increasing with the size to 20 ft. for pipes of 6 in. bore or larger. In this connection it should be noted that the pipe joints should be placed as close to the support as possible.

FORMULÆ RELATING TO WATER, GAS, AND STEAM PIPES

The latest formulæ for water flow in pipes are given below:

No.	*Formula*	*Surface*
I	$V = 47\ m^{.71}\ i^{.57}$	Smooth non-ferrous.
II	$V = 40\ m^{.69}\ i^{.555}$	Bare steel and wrought iron. Asbestos cement. Bitumen lined.
III	$V = 36\ m^{.68}\ i^{.54}$	Bitumen-coated steel. Concrete.
IV	$V = 30\ m^{.67}\ i^{.52}$	Galvanised. Cast iron. Coated cast iron.

It must be emphasised that these formulæ apply to pipes in new condition and with water at normal temperatures.

For the meaning of the symbols see page 381.

GAS, FLOW OF, IN PIPES

$$\text{High Pressure } Q = 0{\cdot}76\sqrt{\frac{(P_1^2 - P_2^2)}{L\ U}\ \frac{d^6}{(d + 1{\cdot}714)}}$$

$$d = \sqrt[5]{\frac{1{\cdot}75\ Q^2\ L\ U}{P_1^2 - P_2^2}} + 0{\cdot}26.$$

$$\text{Low Pressure } Q = 0{\cdot}65\sqrt{\frac{P_1 - P_2\ d^5}{L\ U}}.$$

THICKNESS OF PIPES UNDER INTERNAL PRESSURE

Type of Pipe	*Working Pressure (f.o.s. = 4) lb. per sq. in.*	*Test Pressure ‡ lb. per sq. in.*
Weldless — Low Tensile	$t = \frac{PD}{24{,}300}$ in.	$t = \frac{PD}{38{,}700}$ in.
Weldless — High Tensile	$t = \frac{PD}{35{,}100}$ in.	$t = \frac{PD}{55{,}800}$ in.
Roll Lap-welded*	$t = \frac{PD}{21{,}870}$ in.	$t = \frac{PD}{34{,}830}$ in.
Hydraulic Welded*	$t† = \frac{PD}{24{,}300} + 0{\cdot}047$ in.	$t† = \frac{PD}{38{,}700} + 0{\cdot}047$ in.

* These two formulæ apply only for values of t not exceeding $\frac{3}{8}$ in.
† If t, as calculated, is not less than 0·375 in., it must be increased by 0·016 in.
‡ Maximum test pressure is 1,000 lb. per square inch unless a greater is specially arranged

WEIGHT PER FOOT OF STEEL PIPES:

$$w = 10{\cdot}68148\,(D - t)t$$

Meaning of Symbols

t = thickness of pipe, in inches.
D = outside diameter of pipe, in inches.
d = actual bore of pipe, in inches.
L = length of main, in feet.
h = friction head lost in main, in feet.
V = velocity of flow, in feet per second.
m = hydraulic mean radius in inches.
= $d/4$ for a pipe running full.
i = h/L = hydraulic gradient.
P = pressure, in pounds per square inch.
P_1, P_2 = initial and final absolute pressures, in pounds per square inch.
p_1, p_2 = initial and final absolute pressures, in inches of water.
Q = discharge of gas, in cubic feet per second at atmospheric pressure.
U = specific gravity of gas relative to air.

Expansion of Steam Pipes

The coefficient of expansion of steel steam pipes is given by the formula:

$$a = (6{\cdot}3 + 0{\cdot}0017\,T) \times 10^{-6},$$

where a = average coefficient of linear expansion per degree F. from 0° F.
T = higher limit of temperature in degrees F.

Hence the expansion (e) for a rise of temperature from T_1 °F. to T_2 °F. is given by:

$$e = \frac{T_2 - T_1}{10^6}\left[7{\cdot}56 + 0{\cdot}00204\,(T_2 + T_1)\right] \text{ in. per foot.}$$

For example, for the range 60° F. to 600° F. we have:

$$e = \frac{600 - 60}{100{,}000}\,(7{\cdot}56 + 0{\cdot}00204 \times 660)$$

$$= \frac{540 \times 8{\cdot}9064}{100{,}000} = 0{\cdot}0481 \text{ in. per foot.}$$

The following table gives the value of e for various temperatures, the lower being taken as 60° F. throughout.

Range of Temperature ° F.	60°–250°	60°–300°	60°–350°	60°–400°	60°–450°	60°–500°	60°–550°
Expansion in inches per foot . . .	0·0156	0·0199	0·0243	0·0289	0·0335	0·0383	0·0431

Range of Temperature ° F.	60°–600°	60°–650°	60°–700°	60°–750°	60°–800°	60°–850°	60°–900°
Expansion in inches per foot . . .	0·0481	0·0531	0·0583	0·0636	0·0689	0·0744	0·0799

Flow of Steam in Pipes

In determining the size of a steam main, the known factors are usually the quantity of steam to be conveyed, the initial pressure and temperature, and the length of the main. These are not sufficient definitely to fix the size of the pipe, as the permissible pressure drop or the velocity of flow must first be decided upon. These two factors are interdependent, and both are subject to limits which it is not wise to exceed.

It is not possible to lay down any hard-and-fast rules on the subject of velocities of steam flow, as so much depends on the circumstances of each individual case. In addition to the velocities being limited by the permissible drop in pressure, there are certain maximum values which should not be exceeded. One of the chief limiting factors is the erosive action of the steam on the valve seats and other similarly exposed parts. The velocity of wet steam should therefore not be as high as that of superheated steam.

The following values may be used as a guide:

	Ft. per sec.
For exhaust steam	70–100
For saturated steam	100–130
For superheated steam . . .	130–200

If it is decided to use a definite velocity, the bore of the pipe may be determined from the following formula:

$$d = 1{\cdot}75\sqrt{\frac{Wv}{V}} \qquad \text{(1)},$$

where d = bore of pipe in inches.
W = weight of steam in pounds per minute.
v = mean specific volume of steam in cubic feet per pound.
V = steam velocity in feet per second.

Where the pressure drop and not the velocity of the steam is fixed, the various factors are connected by the equation:

$$W = 87\sqrt{\frac{P_1 - P_2}{vL} \times \frac{d^6}{d + 3{\cdot}6}} \qquad \text{(2)},$$

which gives the rate of flow directly. In this formula
P_1 = initial pressure in pounds per square inch.
P_2 = final pressure in pounds per square inch.
L = equivalent length of pipe in feet.

The "equivalent length" of the pipe is its actual length, plus an allowance sufficient to compensate for the additional friction of any fittings in the installation as compared with straight pipe.

If it is the permissible pressure drop (which is not usually permitted to exceed 2 per cent. of the initial pressure per 100 feet of pipe) that is the determining factor, equation (2) can be written:

$$P_1 - P_2 = 0{\cdot}000132 \left(1 + \frac{3{\cdot}6}{d}\right) \frac{W^2Lv}{d^5} \quad \ldots\ldots\ldots\ldots (3)$$

Frequently, however, it is the bore of the pipe which is to be determined, when equation (2) can be rearranged as:

$$d^6 - \frac{W^2vL}{87^2 (P_1 - P_2)} (d + 3{\cdot}6) = 0 \quad \ldots\ldots\ldots\ldots (4)$$

which can be written:

$$d^6 - Md - 3{\cdot}6\, M = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

where

$$M = \frac{W^2vL}{87^2 (P_1 - P_2)}$$

To determine the diameter of a steam pipe given the weight of steam in pounds per minute, the equivalent length of pipe in feet, and the initial and final pressures in pounds per square inch, the following approximate formula may be used. The value of M must first be found, using the equation above:

$$d = \sqrt[5]{M} \left(\frac{2{\cdot}88 + \sqrt[5]{M}}{2{\cdot}16 + \sqrt[5]{M}}\right) \ldots\ldots\ldots\ldots\ldots (6)$$

Although this formula is not strictly accurate, it will be found to give results which are accurate enough for all practical purposes, provided the bore of the pipe (d) is not less than 3 in.

Range of Temperature ° F.	60°–600°	60°–650°	60°–700°	60°–750°	60°–800°	60°–850°	60°–900°
Expansion in inches per foot . . .	0·0481	0·0531	0·0583	0·0636	0·0689	0·0744	0·0799

Flow of Steam in Pipes

In determining the size of a steam main, the known factors are usually the quantity of steam to be conveyed, the initial pressure and temperature, and the length of the main. These are not sufficient definitely to fix the size of the pipe, as the permissible pressure drop or the velocity of flow must first be decided upon. These two factors are interdependent, and both are subject to limits which it is not wise to exceed.

It is not possible to lay down any hard-and-fast rules on the subject of velocities of steam flow, as so much depends on the circumstances of each individual case. In addition to the velocities being limited by the permissible drop in pressure, there are certain maximum values which should not be exceeded. One of the chief limiting factors is the erosive action of the steam on the valve seats and other similarly exposed parts. The velocity of wet steam should therefore not be as high as that of superheated steam.

The following values may be used as a guide:

	Ft. per sec.
For exhaust steam	70–100
For saturated steam	100–130
For superheated steam . . .	130–200

If it is decided to use a definite velocity, the bore of the pipe may be determined from the following formula:

$$d = 1{\cdot}75\sqrt{\frac{Wv}{V}} \quad \text{(1)},$$

where d = bore of pipe in inches.
W = weight of steam in pounds per minute.
v = mean specific volume of steam in cubic feet per pound.
V = steam velocity in feet per second.

Where the pressure drop and not the velocity of the steam is fixed, the various factors are connected by the equation:

$$W = 87\sqrt{\frac{P_1 - P_2}{vL} \times \frac{d^6}{d + 3{\cdot}6}} \quad \text{(2)},$$

which gives the rate of flow directly. In this formula
P_1 = initial pressure in pounds per square inch.
P_2 = final pressure in pounds per square inch.
L = equivalent length of pipe in feet.

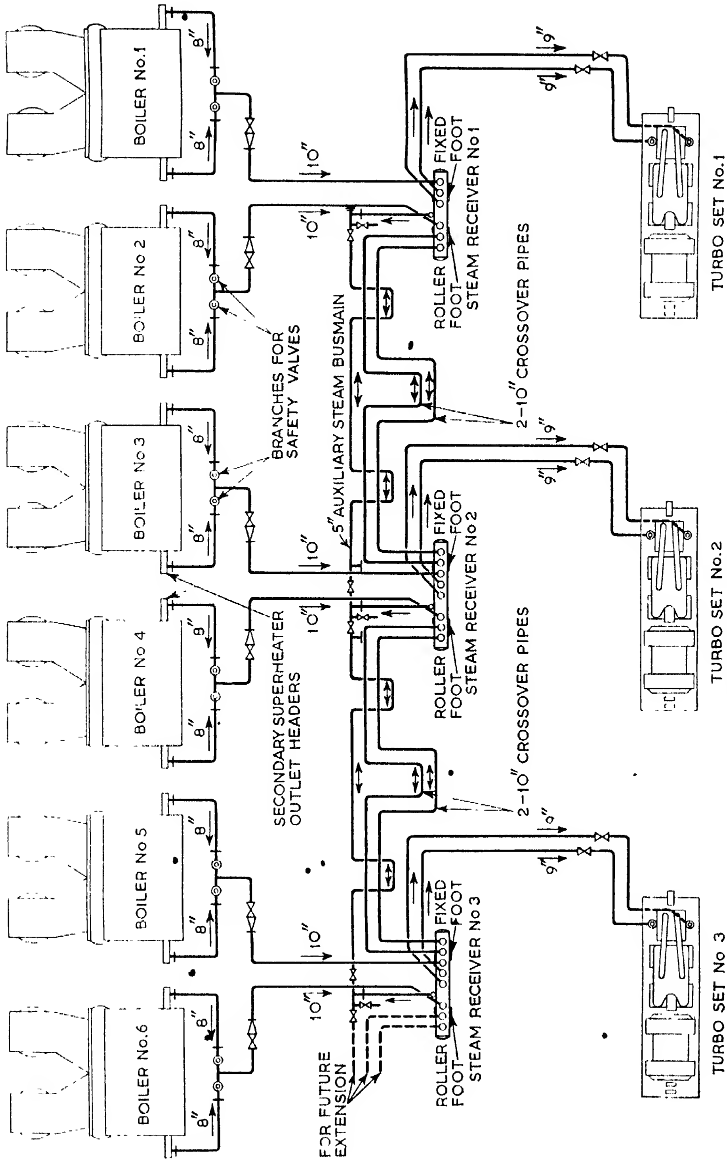


FIG. 1.—STEAM-PIPE DIAGRAM FOR A TYPICAL POWER-PLANT INSTALLATION

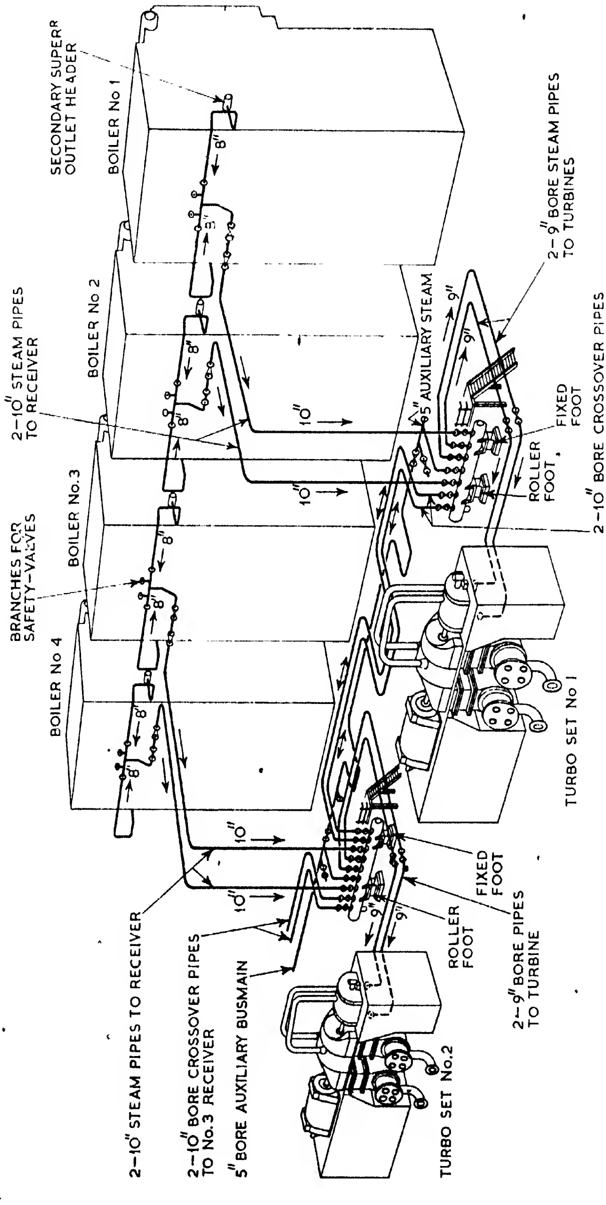


FIG. 2.—LAYOUT OF STEAM PIPING FOR FOUR OF THE BOILERS AND TWO TURBO-ALTERNATORS SHOWN IN FIG. 1

Pipe Bores

From the data the pipe bores are fixed, using a steam velocity of 140–180 ft. per second, at the same time ensuring that the calculated pressure drop in the system is within the allowable figures for the design conditions of the boiler and turbine. Fig. 1 indicates the typical pipe bores for such an installation.

Type of Steel Required

With this installation, the pipework design pressure will be assumed to be 1,030 lb. per square inch, with a temperature at the boiler outlet stop valve of 925° F., and this temperature decides the type of pipe steel which is necessary. A chrome-molybdenum steel having the following chemical and physical analysis is suitable for these working conditions:

Carbon percentage	*Silicon percentage*	*Sulphur percentage*	*Phosphorus percentage*	*Manganese percentage*	*Chromium percentage*	*Molybdenum percentage*
0·15 max.	0·3 max.	0·04 max.	0·04 max.	0·3–0·6	0·7–1·2	0·45–0·6

Ultimate tensile strength : 28–36 tons per square inch (minimum).
Yield point : 50 per cent. ultimate tensile strength (minimum).
Elongation : 25 per cent. on 2 in. (British Standard Test Piece "C") (minimum).

This steel is selected because it has good creep-resisting properties when stressed at the working temperature for long periods. At the same time it remains ductile. It also possesses good forging qualities, and is suitable for drawing into tubes, which satisfactorily withstand the conditions imposed on them during manufacture into finished pipework, by bending, flanging, and by welding. The manufacture of the steel may be in an acid open hearth, or in an electric furnace, but must be closely controlled and deoxidised with a minimum of aluminium, preferably of the order of 8 oz. per ton maximum. The admixture of chromium combined with the minimum of aluminium ensures a steel resistant to graphitisation.

Chromium further helps the steel to be drawn more smoothly and assists in producing a good surface on the tube, both internally and externally.

Calculating Pipe Thickness Required

The next stage is to determine the pipe thickness required. Consideration must, however, be given to the method of manufacture, as the back of the pipe bent by the plain process is thinned a certain amount and this must be allowed for in the calculated pipe thickness. Pipes manufactured by the creased process (Fig. 3) can be fabricated from the same thickness of tube as straight pipes, because no thinning takes place during manipulation. The creases extend for two-thirds of the circumference of the pipe, avoiding the outer wall of the bend. The surplus length on the inner side is absorbed by the creases during the bending of the pipe.

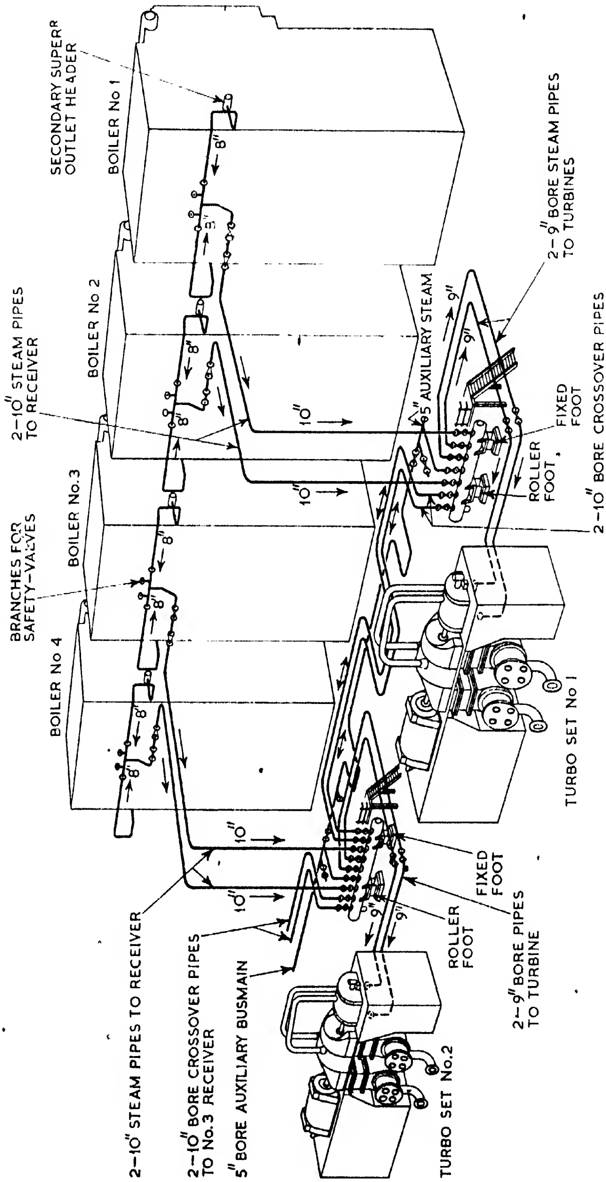


FIG. 2.—LAYOUT OF STEAM PIPING FOR FOUR OF THE BOILERS AND TWO TURBO-ALTERNATORS SHOWN IN FIG. 1

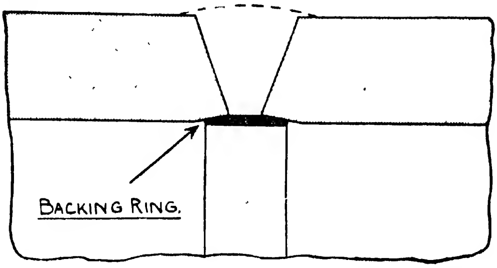


FIG. 4.—END PREPARATION FOR BUTT-WELDING PIPES
(*Aiton & Co., Ltd.*)

pipe anchorages and terminal points are allowable by the boiler and turbine manufacturers at pipe terminals.

A method of calculation is the Grapho-Analytical method as described in the *Piping Handbook* by Sabin Crocker (McGraw-Hill). When making these calculations it is usual to allow the pipes to be pre-stressed cold, by leaving certain gaps in the piping system, known as cold pull-up gaps, which allow this to be done. Particular care must be taken in the choice of these, however, as with certain types of pipe-to-pipe joints, difficulty may be experienced. This will be referred to later.

Another feature of design is shown in Fig. 1, where it will be seen that the steam receiver is anchored. This allows each pipe run to be calculated as an individual problem so that the stresses and thrusts can be guaranteed.

Pipe Jointing

Dealing now with the method of jointing pipe to pipe, pipe to valve, etc., a type of joint is shown in Fig. 4. This is known as a butt-welded joint. The pipe ends or valve ends and the backing ring must be machined to very close tolerances.

The butt-welded joint has been used extensively for working conditions similar to those under review, but it is necessary to have special facilities on site to permit of its construction. Having no connecting flanges with adjoining bolts, as is the case with an ordinary flange joint, it is difficult to erect and line up *in situ*, and special erection equipment is necessary to keep the pipes in line whilst the joint is being erected and subsequently welded.

It will be readily understood that this difficulty is experienced at joints in long vertical mains, and at positions where cold pull-ups are arranged, and this affects the amount of cold pull-up which can be allowed in a butt-welded pipe system. Cold pull-up must be avoided if the piping is of the three-plane variety, as, if the pipe revolves during the making of a particular cold pull-up, the design requirements are not being complied with. It is therefore usual to restrict to a minimum the cold pull-up gaps in butt-welded mains.

Before welding commences the ends of the adjoining pipes must be preheated. This can be successfully carried out by the electric induction method, or by the use of gas, which is more difficult to handle and to control. Metal arc welding, using covered electrodes, is employed. The designer or draughtsman must be careful to position the joints so that the welder and other personnel

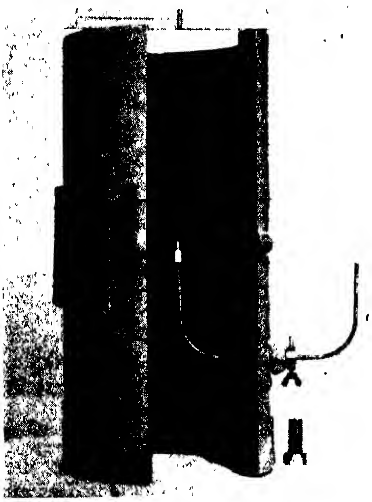


FIG. 5.—PIPE SECTIONED TO DEMONSTRATE THE INSPECTION OF A WELD BY GAMMA RADIOGRAPHY (*Aiton & Co., Ltd.*)

have full access and can be comfortable during the welding and processing of the joint. The work calls for high concentration by the men employed in all stages of manufacture, and if necessary, where a joint is of considerable height above gangways or ground, a special staging must be erected in order to relieve the men of any anxiety during attention to their duties. The welder should be a specialist, trained and tested.

On completion of welding, the joint is stress relieved, calling for further apparatus either in the form of electric induction coils or gas-heated furnaces suitably controlled and recorded.

Radiographic Examination of Pipe Joints

It is usual for each joint to be examined by X-ray or gamma-ray, in order that any flaw which is present in the weld may be detected, cut out, and rewelded as necessary, followed by further X- or gamma-ray inspection. A Welding Supervisor should be present, who is not only capable of passing an opinion on the welding technique, but has metallurgical qualifications, and so is able, not only to take radiographs of the welds, but can interpret and make decisions on the details shown thereon.

It has been found from experience that X-ray equipment is not readily portable to the position of joints in pipework, and Fig. 5 shows a method successfully employed for the taking of gamma-ray photographs.

The source of gamma rays is radium-sulphate in a platinum container, attached by screwing to a wire holder. This source is inserted into the pipe through a screwed boss which is used for fixing the locating device controlling the position of the radium source in the bore of the pipe. The films are placed round the outside of the pipe in suitable cassettes. On completion of gamma-ray examination a plug is inserted into the boss and permanently welded up.

This has proved a very convenient method of radiography, and the time required for long exposures by this method is offset by taking photographs at night, or when the men are not actually engaged on the work.

If, during the fabrication of pipes having ends prepared for butt-welding it is necessary to weld branches or carry out work which calls for hydraulic testing, special arrangements have to be made by plugging the pipes for the hydraulic test at the fabricator's works.

Aiton Patent "Corwel" Joint

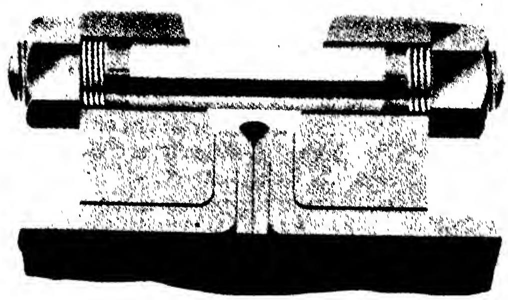


FIG. 6.—ILLUSTRATING THE AITON PATENT "CORWEL" JOINT
(Aiton & Co., Ltd.)

As an alternative to the butt-welded joint, an Aiton Patent "Corwel" joint (Fig. 6) can be employed. This type of joint is constructed on pipes at the maker's works, and erection of the joint can proceed in the same way as with the older method of erecting a packed joint. If manufactured in chrome-molybdenum steel, it requires preheating before the seal welding is carried out, but it does not require any other special technique on site; for instance, stress relieving is not necessary, as the weld is only of the seal type. The weld is supported by loose flanges and bolts, so it is not necessary to subject the weld to X-ray or any other method of examination.

The "Corwel" joint can be successfully adapted to valves, connections from superheater outlet headers, and other terminal connections, enabling the whole of the pipework system to be welded up and to be free from all joint troubles. If occasion demands, the weld can easily be cut out, though of course care is necessary to prevent damage to parent metal. When a joint is broken, the weld is blown out and the welding bevel cleaned by a chisel, file, or portable grinder. Joints may also be broken using an air or hand hammer and chisel, or portable grinder with a suitably shaped wheel to remove the weld metal, after which the Corwel joint may be rewelded and put back into service.

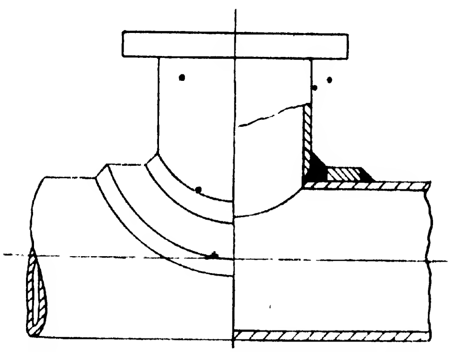


FIG. 7.—TYPE OF REINFORCEMENT NECESSARY FOR WELDED BRANCH CONNECTION *(Aiton & Co., Ltd.)*

The loose-ring flanges will be in chrome-molybdenum steel of similar analysis to the tube material, and the stud-bolts which unite the flanges will be in molybdenum-vanadium steel, having the necessary resistance to creep at high temperature.

Where branches are welded to pipes or receivers, suitable reinforcement is necessary and needs to be fitted. A typical reinforcement is shown in Fig. 7.

Pipe Supports

A pipe-line must receive careful consideration with regard to supporting, particularly where large vertical movements occur due to thermal expansion. Chrome-molybdenum piping, of the analysis previously detailed, expands 8·381 in. per hundred feet at a working temperature of 925° F. It is usual to fit spring-tension supports at all positions of vertical movement. These may be of the straightforward helical-spring type, constant-load spring type, or counter-weight-balance type.

During installation in the power-station the design requirements must be strictly maintained, all drainage falls being checked so that when the pipe expands, up to temperature, the drainage falls are not neutralised.

In the case of a butt-welded pipe-line, it is usual for a hydraulic test to be carried out on the completed installation, but this is not usually necessary when piping is fitted with joints of the flanged type.

After installation, the completed pipe system will be insulated with an approved pipe covering.

The above remarks apply to one specific set of working conditions. A great deal could be written with regard to other working conditions; for instance, carbon steel is usually employed up to a temperature of 850° F. with a maximum of 875° F. Chromium steel, mentioned on page 387, is utilised for temperatures up to 975° F. Over this temperature other steels—molybdenum-vanadium and austenitic are used as required by the specific working conditions.

Corrosion Fatigue

It is essential to point out that corrosion fatigue cracking is liable to occur in any material which is subjected concurrently to sufficiently high fluctuating stresses and the corrosive action of water. Therefore, when once a pipe system is installed, it is necessary to instruct the operators upon the correct operation of main control valves and drain valves, the maintenance of steam traps, and other items of equipment to avoid such an occurrence.

Where the flexibility cannot be obtained by normal arrangement of pipes, corrugated straights or bends can be incorporated.

Conclusion

Whilst this article has dealt with a conventional type of installation, there are many other types of power plants, all calling for special consideration of the piping problems involved.

J. C.

Aiton Patent "Corwel" Joint

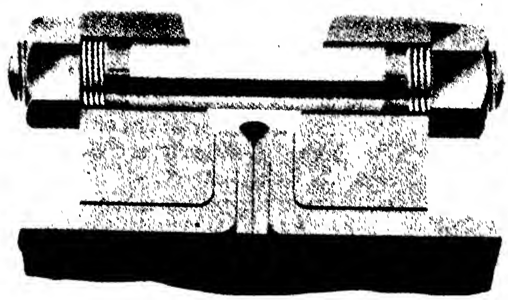


FIG. 6.—ILLUSTRATING THE AITON PATENT "CORWEL" JOINT (*Aiton & Co., Ltd.*)

As an alternative to the butt-welded joint, an Aiton Patent "Corwel" joint (Fig. 6) can be employed. This type of joint is constructed on pipes at the maker's works, and erection of the joint can proceed in the same way as with the older method of erecting a packed joint. If manufactured in chrome-molybdenum steel, it requires preheating before the seal welding is carried out, but it does not require any other special technique on site; for instance, stress relieving is not necessary, as the weld is only of the seal type. The weld is supported by loose flanges and bolts, so it is not necessary to subject the weld to X-ray or any other method of examination.

The "Corwel" joint can be successfully adapted to valves, connections from superheater outlet headers, and other terminal connections, enabling the whole of the pipework system to be welded up and to be free from all joint troubles. If occasion demands, the weld can easily be cut out, though of course care is necessary to prevent damage to parent metal. When a joint is broken, the weld is blown out and the welding bevel cleaned by a chisel, file, or portable grinder. Joints may also be broken using an air or hand hammer and chisel, or portable grinder with a suitably shaped wheel to remove the weld metal, after which the Corwel joint may be rewelded and put back into service.

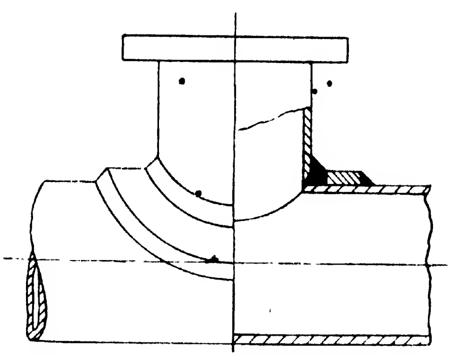


FIG. 7.—TYPE OF REINFORCEMENT NECESSARY FOR WELDED BRANCH CONNECTION (*Aiton & Co., Ltd.*)

The loose-ring flanges will be in chrome-molybdenum steel of similar analysis to the tube material, and the stud-bolts which unite the flanges will be in molybdenum-vanadium steel, having the necessary resistance to creep at high temperature.

Where branches are welded to pipes or receivers, suitable reinforcement is necessary and needs to be fitted. A typical reinforcement is shown in Fig. 7.

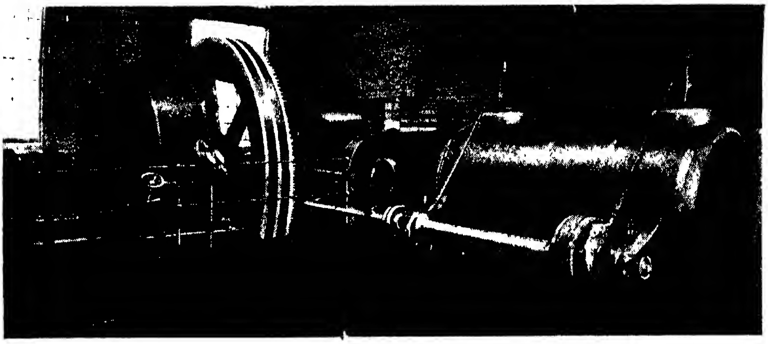


FIG. 2.—A 1,250-H.P. HORIZONTAL UNIFLOW STEAM ENGINE DRIVING A GENERATOR

The Uniflow engine has an inlet valve at each end of the cylinder and no exhaust valves. An automatic relieving gear enables the attendant to change from condensing to non-condensing without having to operate any valves. (*Robey & Co., Ltd.*)

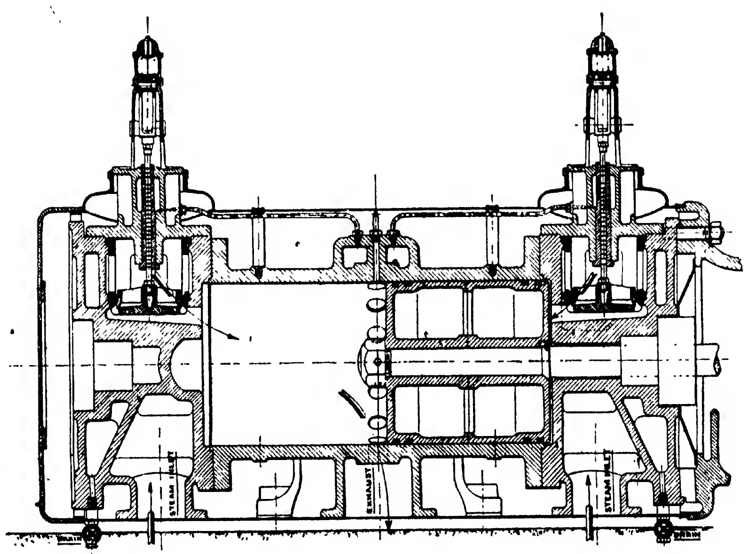


FIG. 2A.—SECTION OF CYLINDER OF UNIFLOW STEAM ENGINE

The steam flows in one direction only, entering through a valve at each end of the cylinder and leaving through ports at the centre which are opened and closed by the piston. Note that the cool exhaust steam does not pass near the hot inlet surface. (*Robey & Co., Ltd.*)

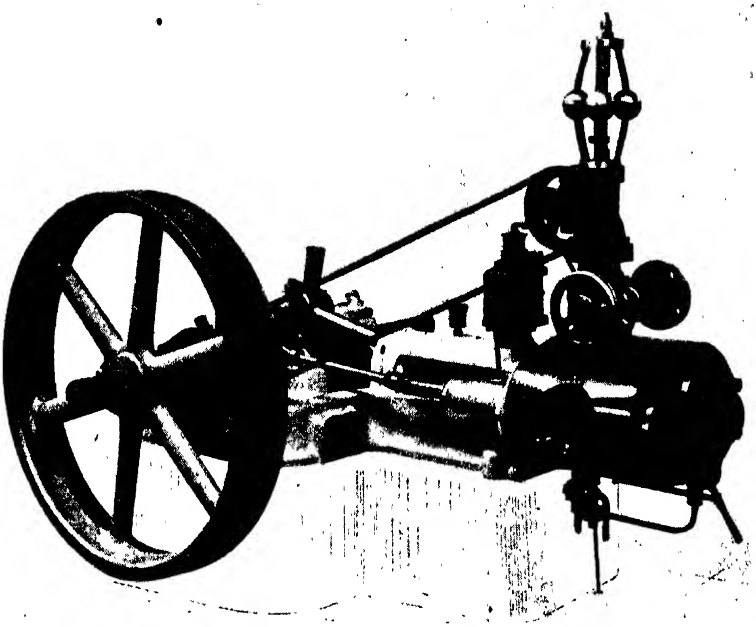


FIG. 3.—HORIZONTAL STEAM ENGINE FITTED WITH PICKERING-TYPE GOVERNOR AND PISTON-VALVE GEAR (*Marshall, Sons & Co., Ltd.*)

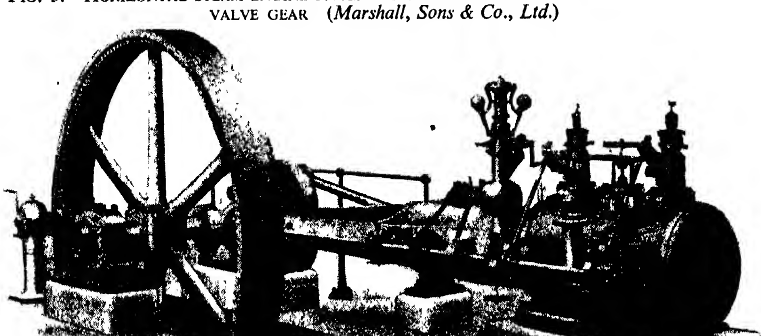


FIG. 4.—A HORIZONTAL ENGINE FITTED WITH TRIP-VALVE GEAR

The trip gear consists of equilibrium double-vent steam and exhaust valves, one of each for each end of the cylinder, with suitable operating mechanism driven from the layshaft, which is geared to the engine crankshaft by mitre wheels. Piston speed up to 600 ft. per minute. (*Marshall, Sons & Co., Ltd.*)

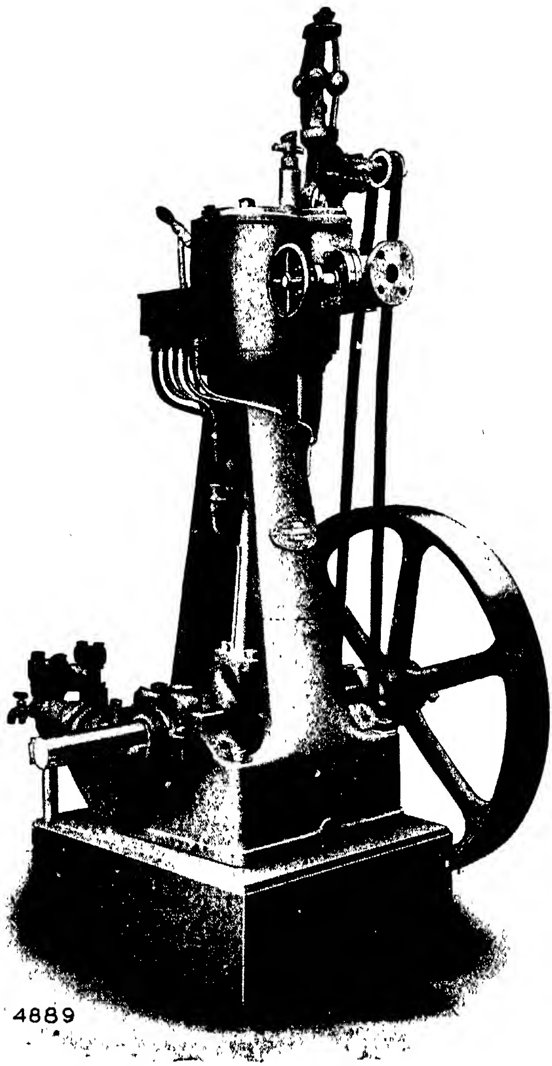


FIG. 5.—VERTICAL SINGLE-CYLINDER SLIDE-VALVE ENGINE WITH PICKERING-TYPE GOVERNOR

Note central lubrication by oil-box and pipes. (*Ransomes, Sims & Jeffries, Ltd.*)

The stationary engines obtain their steam from many types of boilers, economic, Cornish, Lancashire, vertical, water-tube, locomotive type, etc. The portable or combined engines, in the majority of cases, are supplied by locomotive or circular firebox boilers. An article giving a "Survey of Steam Boiler Plant" will be found on page 186.

Stationary Plant

A selection of different types of fixed plant of relatively low speed is shown in Figs. 1–9.

Figs. 3 and 5 illustrate typical simple steam engines, horizontal and vertical, fitted with piston and slide valves respectively and Pickering-type governors.

High-speed engines are generally enclosed, with forced lubrication. Such an engine is shown in

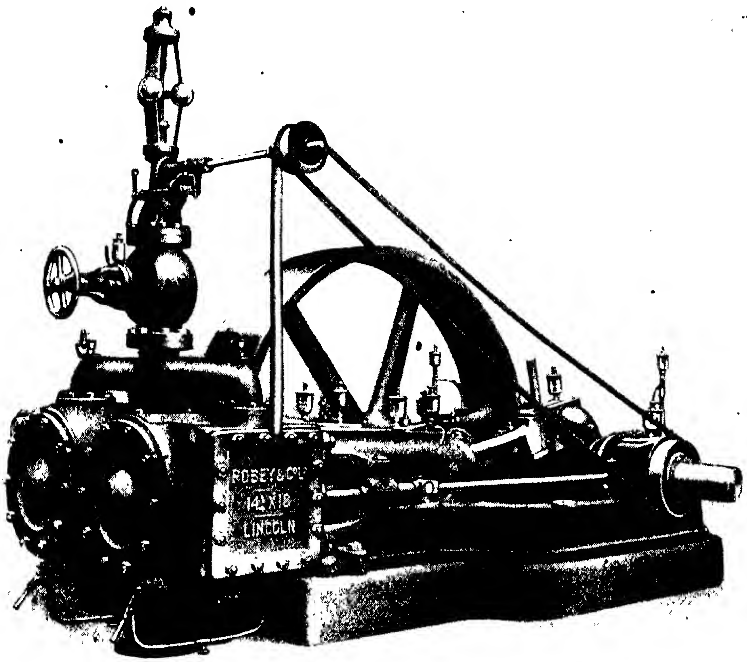


FIG. 6.—A HORIZONTAL DOUBLE-CYLINDER MEDIUM-STROKE STEAM ENGINE

A compact engine giving large power compared with the floor space occupied. (*Robey & Co., Ltd.*)

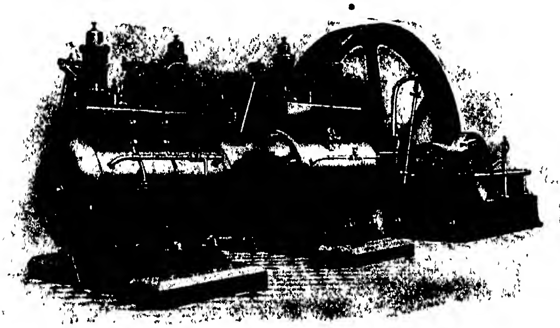


FIG. 7.—HORIZONTAL TANDEM COMPOUND STEAM ENGINE, WITH DROP-VALVE GEAR

Working pressure 160 lb. per square inch. The valves are opened and closed by means of cams mounted in valve bonnets and rocked by a single eccentric. (*Easton & Johnston, Ltd.*)

Fig. 17, and is often coupled direct to a dynamo, the combination being termed a "steam dynamo."

STARTING UP AND RUNNING

Starting up a steam engine involves first of all the raising of steam, and it is assumed that a new plant or a plant which has been some time out of commission is being dealt with.

Manhole and Mudhole Door Joists

The first thing to be done is to make the manhole and mudhole door joints on the boilers (Figs. 10 and 11). These are made with woven asbestos rings well graphited to avoid sticking. The ring should be a good fit on the door. Loose rings are dangerous, as they may get displaced in fitting and blow out. When the ring is on, try the door in its hole, and see that the lip (Fig. 11) enters the hole equally all round. If it does not, remove the door and feel round the inside of the mudhole.

Sometimes fragments of old joint are left adhering to the plate, and unless these are removed, it will be impossible to make a tight joint.

The same remarks apply to the doors, which must be quite clean on the

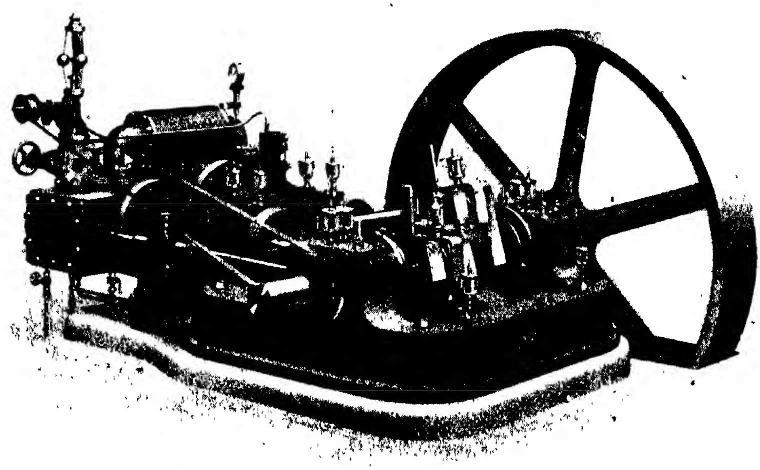


FIG. 8.—COUPLED COMPOUND-TYPE MEDIUM-STROKE HORIZONTAL STEAM ENGINE

The engine illustrated above is fitted with Pickering-type governor. Standard sizes of this type range from 11 b.h.p. to 96 b.h.p. and are suitable for working with steam up to 150 lb. For very narrow sites the cylinders are placed in tandem instead of side to side. (*Robey & Co., Ltd.*)

place where the rings fit. If all is right, put the dog (Fig. 12) on the door stud and then the washer and nut and screw up.

Some doors are of pressed steel, and these are the easiest to apply and the safest in use. The manhole door usually has two dogs (Fig. 10), and these must be done up equally, giving a part turn first to one nut and then to the other.

While the door must be done up tightly, care should be exercised, otherwise the studs or dogs may get broken, and the doors, if of cast iron, broken across the stud (Fig. 13).

It is better to follow up the joints as the boiler heats up, but no extra tension should be put on the studs once there is pressure in the boiler, as in the event of the stud or door breaking, very serious injuries might ensue.

The special woven asbestos joint rings may not always be available, and in such cases a hollow canvas and rubber packing is used. This is cut to length, and a strip of lead rod inserted in each end to form the ring, and the joint is then bound with adhesive tape (Fig. 14). This makes a satisfactory joint, but will not stand being used more than once, as it bakes on to the door or boiler plate and has to be chipped off.

After all the joints are made, turn off the blow-off cock and see that all fittings are in order.

When steam is being raised, the manhole and mudhole doors should be inspected for leakage and tightened, care being taken not to force them.

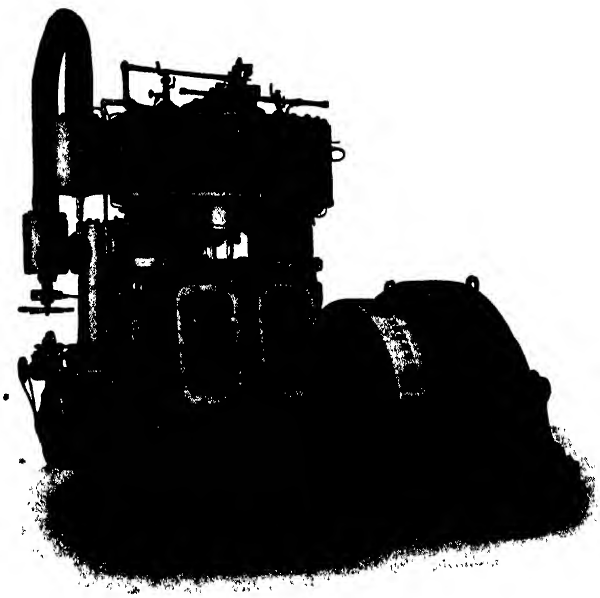


FIG. 9.—COMPOUND INCLINED-VALVE ENCLOSED HIGH-SPEED ENGINE

With slide valves and automatic expansion gear. Direct-coupled to dynamo. Note steam drier on steam pipe. (*Belliss & Morcom, Ltd.*)

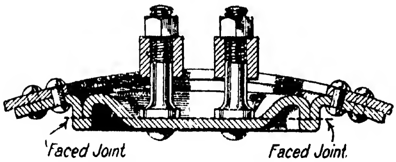


FIG. 10.—PRESSED-STEEL OR McNIELL-TYPE MANHOLE DOOR

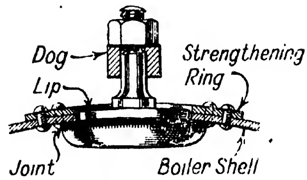


FIG. 11.—CAST-IRON MUDHOLE DOOR

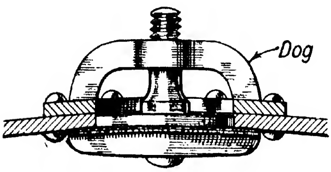


FIG. 12.—DOG SLIPPED ON STUD OF DOOR READY FOR TIGHTENING UP

Dogs are usually forged steel or steel stampings.

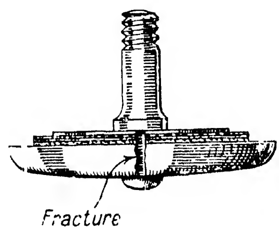


FIG. 13.—CAST-IRON MUDHOLE DOOR FRACTURED ACROSS STUD HOLE DUE TO EXCESSIVE TIGHTENING OR UNEQUAL BEDDING OF JOINT

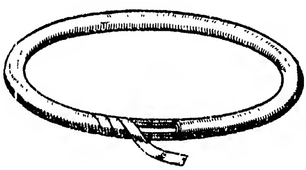


FIG. 14.—HOLLOW GASKET TUBING OF RUBBER AND CANVAS

Can be made up to any size door. Note lead plug at joint, which is afterwards wound with adhesive tape.

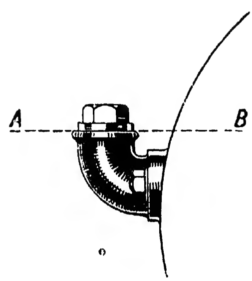


FIG. 15.—FILLING PLUG ON BOILER

Usually arranged so that boiler cannot be overfilled. *AB*, maximum height of water.

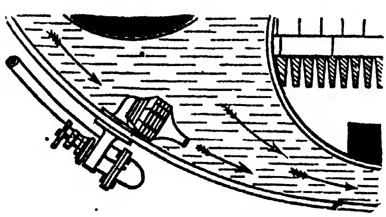


FIG. 16.—CIRCULATOR

For circulating the "dead" water in boilers and thus avoid danger of straining.

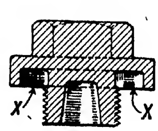


FIG. 15A. — SECTION OF BOILER FILLER PLUG

Note recess for joint *XX*, to prevent it being blown out.

Attention to Gauges

See that the gauge glasses are sound and that the rubber packing rings are fitted. Fill the siphon of the pressure gauge with water, and if there is a cock in the gauge siphon, see that it is turned on.

Firebars

If the firebars have been removed, these should be replaced and care taken to see that they are equally spaced. Bent or defective bars cause immense waste of fuel and should be discarded; the bars should be clean on top. If burned away, new bars should be fitted. New bars are cheaper than a loss of even ½ per cent. of fuel, and defective bars can easily waste 20 per cent.

Filling the Boiler

Fill the boiler with water until it is about half-way up the gauge glass. In boilers of the locomotive type, a filler plug is generally provided in the manhole door, or a separate bend, with a plug, is fitted to the boiler, and in such a position that the boiler cannot be filled above the top of the bend (Fig. 15). A special funnel is generally provided with the engine to enable the water to be poured in conveniently.

On large stationary boilers water can generally be run in from the main by a special connection fitted with a check valve to prevent the boiler pressure getting back into the main in the event of the filling cock being opened while the boiler was under steam, or the water may be supplied from overhead tanks with the same precautions adopted. If the boiler is one of a range and the other boilers are working, it can be filled through the feed check valve in the usual manner.

Filling Plugs

In the cases where filling plugs have been removed, these should be screwed up tightly. The plugs are usually arranged with a recess under the head to prevent the joint being forced out (Fig. 15A), and lead washers are often fitted, or asbestos cord wound into the groove.

Lighting the Fire

Before lighting the fire, see that all valves are turned off, and that the dampers on those boilers which have them are free to work easily. If the boiler is fitted with economisers, care should be taken to see that these are full of water, and that the by-pass damper is open while steam is being raised.

The method of lighting the fire differs with different types of boiler and conditions.

LIGHTING FIRE OF PORTABLE PLANTS.—With ploughing engines, traction, and portable engines in the field, where wood, paper, shavings, etc., are not generally available, knobs of coal about the size of an apple are placed on the bars and bundles of straw are stuffed into the ashpan and lighted, and the coal is soon ignited.

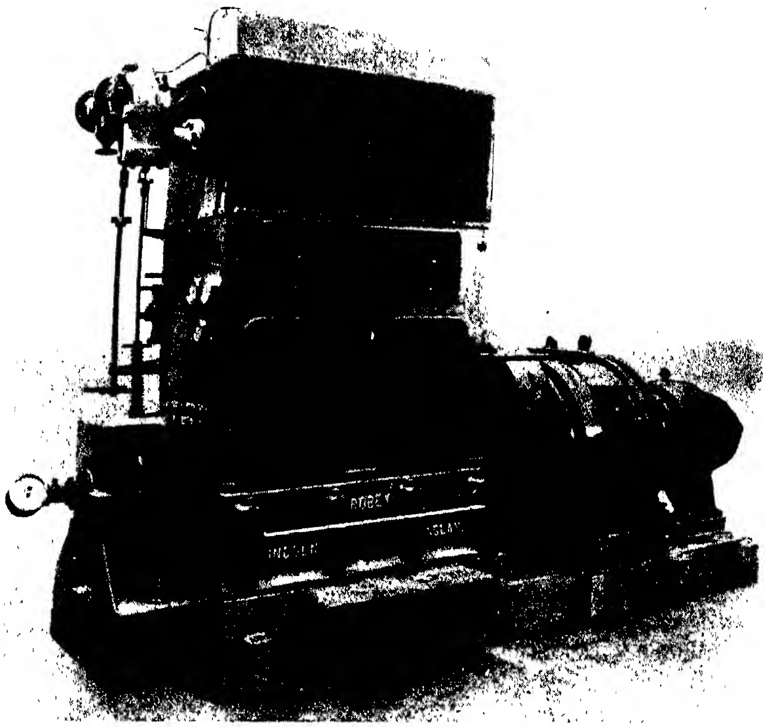


FIG. 17.—A HIGH-SPEED VERTICAL ENGINE OF THE COMPOUND-ENCLOSED TYPE

This engine is fitted with a crankshaft throttle governor, enabling the speed of the engine to be varied whilst running. (Robey & Co., Ltd.)

LIGHTING BOILERS WITH LONG GRATES.—With large boilers with long grates, such as Lancashire, Cornish, etc., the back part of the grates is covered with coal. A small fire is started in the mouth of the furnace with some wood and oily rags covered with small knobs of coal, and the door being shut, the draught soon carries the fire down the furnace tube to the other coal.

Where many boilers are in use, or in locomotive sheds, the fires are generally started with several shovelfuls of live coals out of a large brazier or "devil," and coal fed on to them.

Raising the Steam

Once started, the fire should be kept thin and even on the bars: a thick fire produces heavy smoke and little steam. Fresh coal in square firebox boilers should be fed round the sides and not thrown into a heap in the centre.

In long furnaces the fresh coal should be kept at the door end, so that the gases have to pass over a bed of hot fuel and are thoroughly consumed, instead of passing out in smoke, the fire being pushed forward at intervals to keep a good bed at the back. In this type of furnace a thicker bed of fire has to be carried than on the locomotive type, *especially at the back end.*

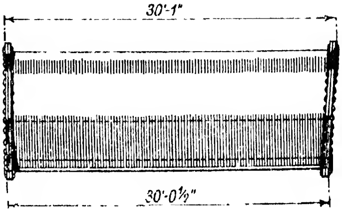


FIG. 18.—BOILER DISTORTED BY TOO *RAPID STEAM RAISING*

Distortion shown exaggerated.

Steam raising should not be hurried from all cold, especially in large boilers containing great quantities of water, or the boilers may be severely strained and seam leakage result.

CIRCULATORS FOR RAISING STEAM.—In Lancashire and marine boilers of the Scotch type, it is not unusual to have nearly full steam pressure at the top of the boiler and almost-cold water at the bottom. To avoid this, circulators are fitted (Fig. 16), in some cases where steam is required rapidly and the danger of leakage is avoided but otherwise it will be realised that a boiler under such conditions may be distorted, as in Fig. 18, to the extent of probably ½ in. in its length.

In all such boilers steam should be raised as slowly as the circumstances permit to allow the brickwork of flues, setting, etc., to warm up gradually. Boilers of the water-tube type are rapid steamers, and few precautions are necessary, as the quantity of water contained is small and the circulation is rapid.

STEAM BLAST IN LOCOMOTIVE BOILERS.—Locomotive-type boilers are rapid steamers, and are generally fitted with a steam blast up the funnel to draw up the fire, but this must be used with discretion, or the tubes will leak at the firebox end. As the steam rises, the height of water in the gauge glasses will rise, due to expansion.

The level in the gauge glass should fluctuate, it should never be steady while the boiler is under steam; a steady level is never a true one, pointing to restricted or choked water or steam passages.

Testing Water Level

Test the level frequently while raising steam by opening the lower cock of the water gauge, and note that the water returns to the same level when closed.

When steam is nearly up to full pressure, close the dampers slightly, or close the ashpan and open the fire door slightly to prevent blowing-off. With tubular boilers, like the locomotive type, the fire door should not be opened long with a fierce fire, as the cold air entering might cause leakage of the tubes owing to rapid contraction.

Draining Water from System

Before turning on steam, it should be ascertained that the drain taps on the main steam pipe, if fitted, are wide open. If steam traps are fitted, there will

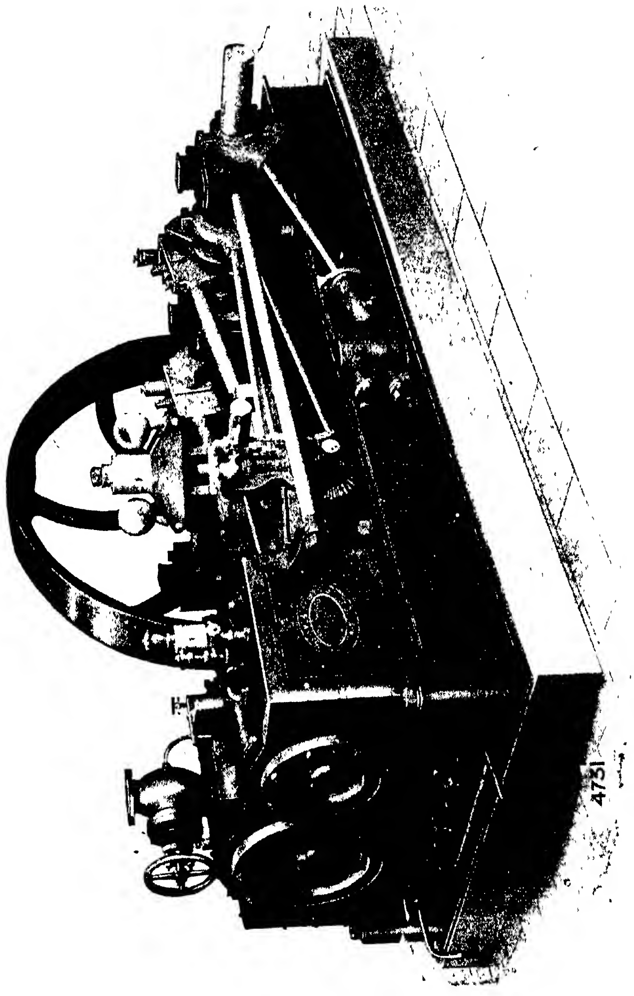


Fig. 19.—Compound horizontal engine

Note drainpipe under seat of stop valve. This keeps the steam pipe drained and supplies steam to the cylinder jackets. The jackets are kept clear of water by a steam trap. The simpling valve for admitting live steam to the L.P. cylinder for starting is shown on the top of the H.P. steam chest. (*Ransomes, Sims & Jeffries, Ltd.*)

be no danger of the steam pipe not being cleared of condensed steam. Open the stop valve very slowly, allowing the steam pipe to warm up gradually. Rapid opening of the stop valve is dangerous, even with steam traps on the pipe, as water hammer might develop and the pipe be fractured.

There is often a drain tap on the main engine stop valve, just under the seat (Fig. 19), and this should be left open until the engine is started. With combined engines and boilers, the cylinder is surrounded by a steam jacket which is open to the boiler, so that the cylinder is hot enough to prevent undue condensation without any blowing through.

Pilot Valves

In large boilers, or in ranges of boilers, pilot valves are often fitted to the main stop valves, and the pilot valve should be opened to warm up the steam main before the main stop valve is opened, thus avoiding any danger from water hammer.

Feed Pump

If there is a separate feed pump apart from any on the engine, test this and see that it delivers water into the boiler satisfactorily. If there is an injector, test this also.

Safety Valve Pressure

If the boiler has not been in use before, or laid up for some time, allow the pressure to rise until the safety valve lifts, and note that the pressure at which it is supposed to blow corresponds with that shown on the pressure gauge.

Heating Up Cylinder before Starting

It is now time to turn to the engine. Before a steam engine is started, it is advisable to make sure that the heat of the cylinder casting is equal to that of the steam. Otherwise the steam entering would condense rapidly, and the water, being forced out by the piston on the exhaust stroke, will set up a heavy pounding, and may, in extreme cases, knock off the cylinder covers.

With slide-valve engines the water will generally escape by forcing the valve off its face without damage, but in the case of piston-valve engines this would not be possible, and relief valves are provided to allow any water to escape without damage. These are really small, spring-loaded safety valves, one arranged in each end of the cylinder (Fig. 20).

It is, however, better to avoid this trouble by heating up the cylinder thoroughly before starting.

Drain Taps on Cylinder

The drain tap on each end of the cylinder should be opened, and also the drain on the steam chest, if one is fitted. If a steam trap is fitted on the steam chest, this should need no attention (Fig. 19). If the engine has jacketed cylinders, a steam trap, in the case of stationary independent engines, is almost

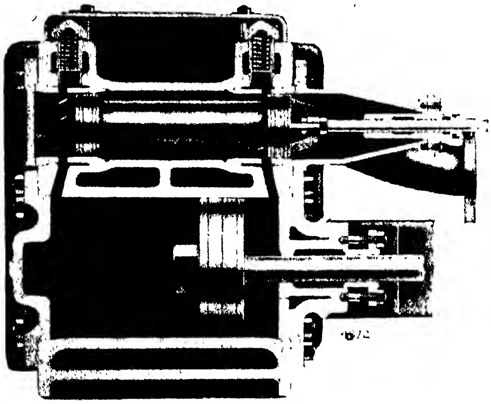


FIG. 20.—SECTION OF CYLINDER FITTED WITH PISTON VALVE

Note spring-loaded relief valves to allow condensed steam to escape. It is impossible to force the piston valve off its seat as with a slide valve. (*Ransomes, Sims & Jeffries, Ltd.*)

invariably fitted. Open the stop valve on the engine very gently and allow steam to flow through the drain tap at the end of the cylinder which is taking steam. The stop valve should then be closed and the engine turned or barred over until the other end of the cylinder is taking steam. Open the stop valve and allow steam to blow through this end.

Simpling Valve

If the engine is compound, a simpling valve is fitted to allow high-pressure steam to enter the low-pressure cylinder for starting purposes should the high-pressure crank stop on dead centres. One of these valves will be seen in Fig. 19. This should be opened when warming up, or the low-pressure cylinder will remain cold, the drain taps on the low-pressure cylinder also being opened. After a few minutes in the case of large engines, the difference in the sound will indicate that steam, not water, is blowing out of the drain taps.

Starting the Engine

SINGLE CYLINDER.—The crank, if it is a single-cylinder engine, should now be placed just over the dead centre, in the direction of rotation, so that the engine can get away.

COMPOUND ENGINE.—If it is a compound engine, the simpling valve should be closed, but all drain taps in both types of engine should be kept open. The stop valve should gently be opened until the engine moves off. In the case of the compound engine it may be necessary to open the simpling valve to admit steam direct to the low-pressure cylinder, and this should be closed as soon as the engine gets under way. As soon as the engine has attained its normal speed and is under the governor control, the drain taps may be closed; the steam stop valve being opened wide.

Maintaining Boiler Pressure

The boiler now requires attention. For economical running the steam pressure should be maintained as even as possible and just under the blowing-off point. If the boiler is oil fired or has a mechanical stoker, little attention will

be required once the plant has settled down to regular work and the burner, or coal feed, is regulated.

With hand firing, however, considerable skill is required in the stoking of the boiler to maintain an even head of steam and to avoid smoke.

Little and often is the rule in stoking. The fuel should be added in small even quantities, and the bars kept free from ash and clinker by regular use of the pick and rake.

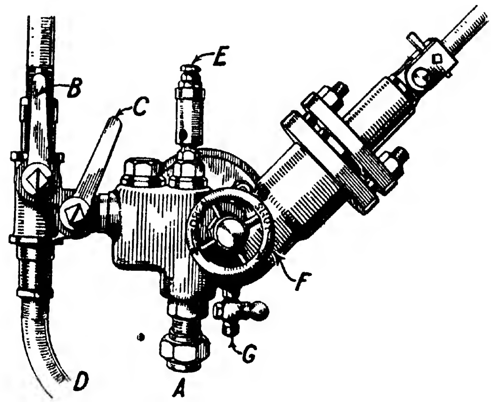


FIG. 21.—FEED PUMP WITH FEED BY-PASS AND HEATING DEVICE

A, suction; *B*, valve controlling steam from main exhaust pipe; *C*, water by-pass valve; *D*, return pipe to feed tank. The exhaust steam mixes with the surplus water on its way to the tank and heats it; *E*, spring relief valve; *F*, valve for shutting off pump from boiler under steam to enable suction and delivery valves to be examined; *G*, air cock.

Locomotive Boilers

In boilers of the locomotive type the fuel should be added evenly at the corners of the firebox and under the door, keeping the centre clear and bright, and also under the tubes in the tubeplate. The fuel, as it gets well alight, can then be moved into the centre to keep the bars from being uncovered, a fatal defect in maintaining steam. A thin fire should be maintained. To stoke heavy charges of fuel and to build up a thick fire merely results in thick smoke, no heat, and a falling steam pressure, the waste of fuel also being enormous. By bad stoking alone the consumption of fuel per horse-power per hour can be increased 50 per cent.

Keep Water-level Constant

It is also essential that the water-level should be kept constant, the feed being so managed to attain this. Engines fitted with feed pumps driven direct have a by-pass fitted on the pump delivery. By regulating this the level of water can be maintained exactly if the load is constant (Fig. 21). To allow the water-level to fall unduly and then run the level up rapidly renders it impossible to maintain a steady pressure, and tends to strain the boiler if the feed is cold.

Boilers with independent feed have feed-regulating valves fitted by which the feed can be very finely adjusted. In such cases the feed is generally hot, the water passing first either through an exhaust steam feed-water heater or through economisers fitted in the flue of the boiler.

Exhaust Steam Feed-water Heaters

Many engines are fitted with exhaust steam feed-water heaters, and it is essential that the tubes of these should be cleaned periodically to remove "fur," or scale. Provision is made for this by fitting detachable cover plates, or ends, giving access to the tubes.

Many combined engines and boilers have an arrangement for diverting part of the exhaust into the pump by-pass, so that the surplus water is heated on its return to the feed tank (Fig. 21).

This has two drawbacks, however: the oil used to lubricate the cylinder passes out with the exhaust steam and finds its way into the feed tank, where it becomes partially emulsified with the feed water and enters the boiler and may cause serious damage due to overheating, as well as the less serious trouble of foaming or priming.

The other trouble is that on periods of heavy loads the feed water in the tank may, unless the heating device is closely watched, get so hot that the pump will not lift it and the feed ceases. Unless some other method of feed is fitted, such as an injector with a cold supply, this may lead to a dangerous situation, as once the pump body has got hot it is difficult to get it to restart, and the water in the feed tank must be brought down to a temperature at which the pump can handle it, say, not over 150° F. Should this situation arise, the fire should be dropped or damped down to avoid melting the fusible plug unless the water gauge reading is fairly high. Some special points dealing with the running of the various types of engines and boilers are dealt with below.

RUNNING OF STATIONARY PLANTS

Operation of Condensing Plants

Many stationary engines are fitted with condensers either of the surface or jet type (Fig. 22). This, where ample water is available, results in considerable economy in working, as the vacuum obtained by the condensation of the exhaust steam is equivalent to an addition of up to 14 lb. per square inch working pressure. All the large turbo-electric generating plants are fitted with condensing plants and, unless a river is available, cooling plants, to cool the condensing or circulating water. It is preferred, however, to limit these instructions to reciprocating engines only.

In some cases the circulating and air pumps are driven direct from the engine; in other cases independent air circulating and feed pumps are used.

On the exhaust pipes of condensing engines a double-seat valve is generally fitted, and this will be seen in Fig. 22. This allows the exhaust steam to be diverted into the open air or into the condenser, as desired.

In the case of engines where the circulating pump and air pump are direct driven, the exhaust steam should be diverted to the air at starting. When the engine is under way and the circulating water has reached the condenser, the exhaust valve is closed to the air and the steam diverted to the condenser.

Jet Condensers.—In the case of jet condensers (Fig. 22), the quantity of

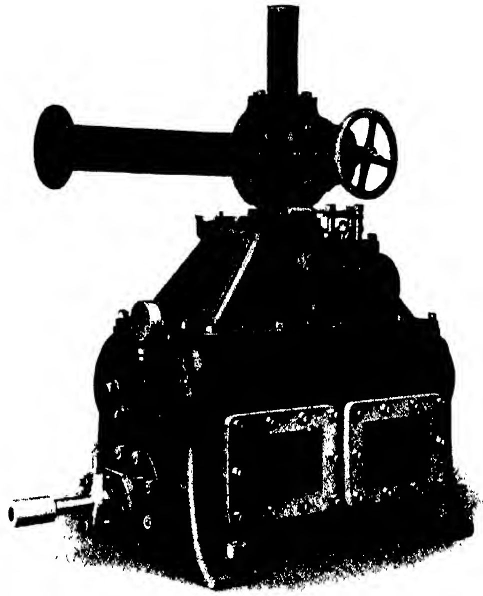


FIG. 22.—JET CONDENSER

The double-seat valve is shown above; the top pipe allows the exhaust steam to escape to the air. Main exhaust pipe on left. The injection water-control valve is shown in front. Vacuum gauge on left. Pump rod and sleeve coupling at bottom, left. (*Ransomes, Sims & Jeffries, Ltd.*)

water should be adjusted until the vacuum is steadily maintained at the highest figure attainable, which will depend on several factors, such as the temperature of the cooling water, the state of the air-pump plunger and valves, tightness of the air-pump glands, etc.

Running of Independent Pumps

Where independent pumps are fitted, the pump is started before the main engine and the exhaust can be turned straight into the condenser. The speed of the independent pumps should be regulated to a minimum necessary to maintain the highest vacuum. In most cases the exhaust of the pump is taken into the main condenser and the pumps are run much too fast. In many installations the auxiliary plant is responsible for great steam, and, consequently, fuel waste.

Steam Driers

On high-speed stationary engines, a steam separator or drier is often fitted on the main steam pipe at the engine to prevent any excess of condensed steam being carried into the cylinders. As these engines are generally fitted with piston valves, it is important that the steam should be as dry as possible for reasons already given. A steam drier will be seen in Fig. 23. It is fitted with a

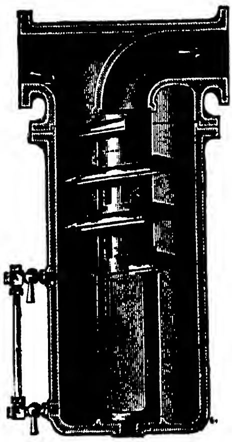


FIG. 23.—STEAM DRIER AS FITTED TO HIGH-SPEED ENGINES

A steam trap is fitted at bottom to keep drier drained. The water gauge shows any accumulation of water. The action is a centrifugal one, the water being thrown against the side of the drier and running to the chamber below.

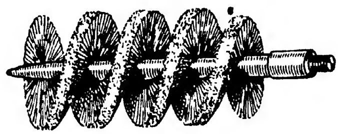


FIG. 24.—SPIRAL WIRE BRUSH FOR BOILER SMOKE TUBES

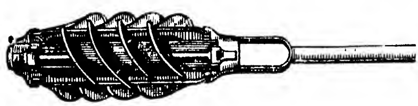


FIG. 25.—SELF-EXPANDING SCRAPER BRUSH

water gauge of the same type as used on steam boilers to indicate any accumulation of water. A drain cock is fitted to the drier and a steam trap to keep the drier free of water is also fitted. The drier needs no attention beyond seeing that the steam trap keeps it free of water.

RUNNING OF COMBINED ENGINES AND BOILERS

In practically all these engines the cylinder is steam-jacketed, the jacket (Fig. 20) being in direct communication with the boiler, so that the cylinder is maintained at the full temperature of the steam and the use of the drain cocks for a few seconds only at starting is sufficient.

GRATE AND ASHPAN.—The grate area of the boiler is relatively small, and it is therefore important to keep the grate clean and free from clinkers and the ashpan clear. Accumulations of ash in the pan cuts off the air to the fire and prevents steam being maintained.

BOILER TUBES.—The boiler tubes also must be kept clean and should be swept daily, a brush of the type shown in Fig. 24 should be used, with a scraper brush (Fig. 25) at weekly intervals.

SMOKE-BOX.—The smoke-box must also be kept clear of soot and ash. This tends to accumulate in the box, being drawn through the tubes by the blast of the exhaust. In a short time it will build up above the ends of the lower tubes, thus stopping the draught, and the ash may catch fire, the lower part of the smoke-box getting red-hot. This can be frequently seen on railway locomotives after a long, heavy run (Fig. 26).

Cleaning-out Locomotive Boilers

Locomotive-type boilers have very restricted waterways and spaces between the tubes. It is essential that soft water should be used as feed. The boiler also must be washed out frequently, all mudhole doors being removed for this purpose, to remove all mud and deposit. The period will vary with the work and the water used as feed, but about 100 hours should be the limit.

Traction, Threshing, and Ploughing Engines—Water-level

In traction, threshing, and ploughing engines, which run on the roads, the water-level on the boiler must receive special attention. While travelling the water-level should be maintained sufficiently high to prevent the crown of the fire-box being uncovered should the engine have to descend a hill (Fig. 27). If there is any doubt, the engine should be stopped before descending the hill and the water-level raised to a safe point. It is best to do this with the injector, as it is not safe to put the engine out of gear if it is on a slope with a heavy load, the engine furnishing the most powerful brake.

For this reason the gear should never, if it can be avoided, be changed on a hill, otherwise the control of the engine and its load may be lost. If there is any doubt of the engine being able to ascend the hill on the high gear, it should be stopped and the gear changed

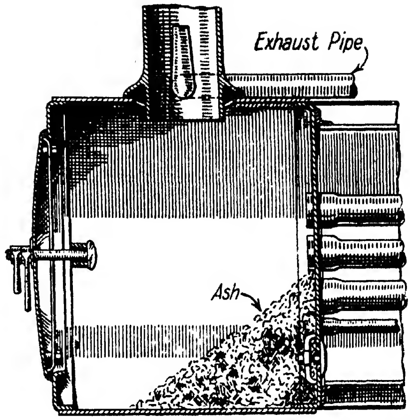


FIG. 26.—SECTION OF SMOKE-BOX OF LOCO-TYPE BOILER

Showing accumulation of fine ash choking bottom rows of tubes.

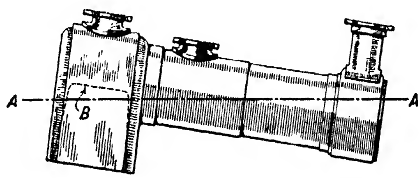


FIG. 27.—WATER-LEVEL IN TRACTION ENGINES

In descending hills, sufficient water must be carried in boiler to avoid top of fire-box becoming uncovered. *AA*, water-level; *B*, firebox.

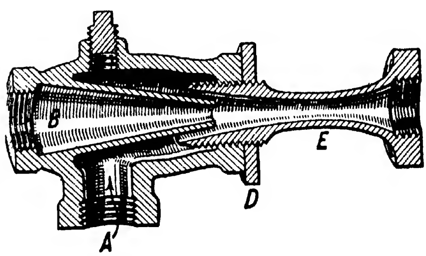


FIG. 28.—STEAM WATER LIFTER FITTED TO ROAD ENGINES AND WAGONS

A, suction; *B*, steam; *E*, delivery to tank; *D*, lock-nut to adjust position of cones.

before the ascent commences. Otherwise all brakes should be applied and the wheels of the whole train scotched or blocked before the change is made.

Picking up Water

All road engines are fitted with a water-lifting device to enable water to be picked up on the road. This is a simple form of steam jet pump (Fig. 28), and will lift water 20 ft. or more and deliver it into the engine tanks. If the water supply is shallow and muddy or gritty, the suction rose on the end of the lifting hose should be laid on the fire shovel placed on the ground to prevent mud, etc., being picked up.

Cleaning-out Water Tank

As any water available has to be used, it is essential that the water tanks should be cleaned out periodically, as otherwise the sediment may build up above the suction openings and so stop the feed to the pump and injector. Mudholes or cover plates are provided on the tanks for this purpose.

Traction Engine driving Machinery

When the engine is driving machinery by belt, it is advisable to place it so that the smoke-box end of the boiler is higher. This ensures drier steam, and allows of a lower water-level to be carried than if the engine was level. The cylinders of these engines are attached to, and take their steam from, the front end of the boiler.

Portable Engines

These engines are used for belt driving only and the cylinders are attached to the rear end of the boiler.

The engine should be kept as level as possible when working, as if inclined, as in the case of road engines, priming may result. These engines, especially those with single cylinders, have a tendency, when working, to rock due to the motion of the reciprocating parts of the engine. Bolted scotches are generally used to hold the engines steady.

Engines of this type are often converted into semi-portable engines by removing the wheels, fitting an ashpan foundation, and carrying the front-end of the boiler on a feed tank.

If part of the exhaust steam is used to heat the feed, this tank gets very foul from the spent lubricating oil deposited in it from the exhaust steam and is, from its position, very difficult to clean. It is advisable to fit a separate feed tank at the side of the engine under the pump and simply let the other tank serve the purpose of a support.

Boiler Feed Pump.—Portable steam engines are generally only fitted with one method of boiler feed, a pump driven off the crankshaft by an eccentric. The pump seldom gives trouble if properly packed and reasonably clean water is used, but it is advisable to fit an alternative method of feeding. Both pumps

and water injectors are almost invariably fitted on traction and threshing engines: they are seldom found on portable engines.

FITTING AN INJECTOR.—If an injector is fitted, a separate feed tank should be provided, as the tank for the pump feed may contain water too hot for the injector to handle. Occasionally the pump fails from this cause, and the injector, with its cold feed water, is then able to carry on the work.

Device for Repairing Engine while under Steam

Provision is made on portable engine pumps for shutting off the pump from the boiler so that in the event of trouble the valves can be removed and examined while the boiler is under steam. This device is shown in Fig. 21 and, as there would be a danger of breaking the pump if the engine was ever started with the pump shut off from the boiler, a spring relief valve is provided (Fig. 21) to avoid damage. When the pump is shut off, the valves can be removed and cleaned or ground in, or the gland of the plunger packed without letting down steam.

Cleaning Water-feed Pipe

When hard water is used, or water containing much sediment, the passage from the pump to the boiler is liable to become scaled up. A plug is provided, closing this passage at the outer end, and by removing the plug a drill can be put through the passage into the boiler and the passage cleared.

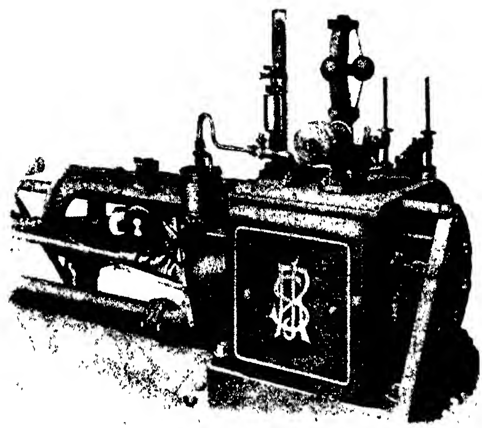


FIG. 29.—MECHANICAL OIL PUMP FITTED TO CYLINDER OF PORTABLE STEAM ENGINE (*Ransome, Sims & Jeffries, Ltd.*)

Protecting Engines in the Open Air

In the case of all engines working in the open air it is essential that a cover should be kept over the top of the chimney when the engine is not working. Otherwise rain falling down the chimney mixes with the soot in the smoke-box and rapidly corrodes the plates of the smoke-box, the ends of the tubes, and the front tube plate of the boiler.

Traction engines are generally supplied with a cap for the chimney. Portable engines, when the chimney is erected, are in some cases protected by a spark catcher, but when the chimney is down, a plate should be put over the chimney base to keep out moisture.

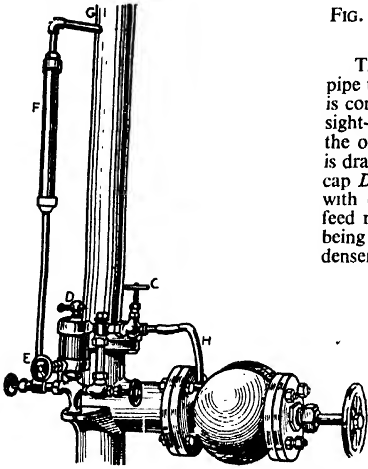


FIG. 30.—APPLICATION OF DISPLACEMENT CYLINDER LUBRICATOR

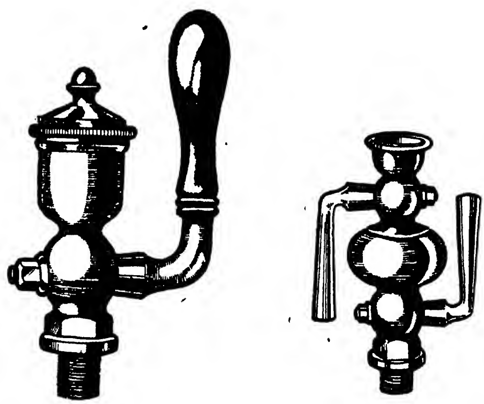
The valve *C* is connected with the steam pipe to engine by the pipe *H*. The lubricator is connected with the steam main at *G*. The sight-glass is filled with water. Valves *C* and the one on the left being closed, any water is drained off at *E*, which is then closed. The cap *D* is removed and lubricator body filled with oil. The cap is then replaced and the feed regulated by valve on left, the valve *C* being kept wide open while working. *F*, condenser. *H*, outlet pipe.

When the engines have finished work for the day, a tarpaulin cover should always be put over the engine. This is supplied by the makers. A very small amount of rain or wet mist will cause serious rusting of all the bright parts of the engine, and apart from the appearance, may damage the working parts.

CYLINDER LUBRICATION

Although the speed of steam engines is relatively slow compared with internal-combustion engines, the bearing pressures are generally higher, and satisfactory running and freedom from breakdown and repairs can only be maintained by paying scrupulous attention to the lubrication.

The lubrication of the cylinder is the first point to consider.



FIGS. 31 AND 32.—SINGLE- AND DOUBLE-CUP CYLINDER LUBRICATORS

Suitable for engines running for short periods only, unless frequently refilled. No effective control of lubrication is possible with these types.

Mechanical Lubricators for Cylinders

With the use of high pressures and superheated steam, more positive methods of lubrication became imperative, and mechanical lubricators or oil pumps are now used for cylinder lubrication on nearly all modern

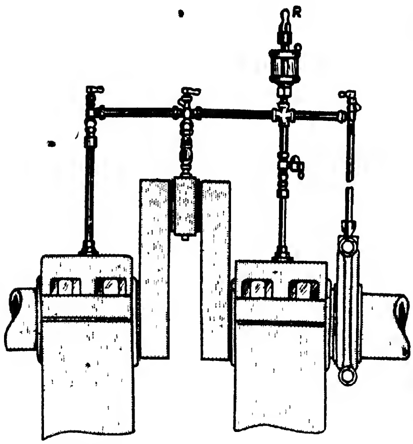


FIG. 33.—SHOWING WIPER CUP FITTED TO CONNECTING-ROD BIG END

The reservoir *R* supplies sight-drip feeds to both main bearings, the eccentric, and connecting rod.

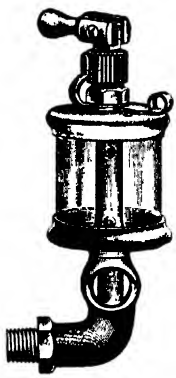


FIG. 34.—LUBRICATOR OF THE SIGHT-FEED TYPE

To set the feed, lever is raised and milled nut is turned until desired feed is obtained.

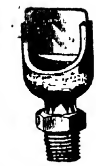


FIG. 35.—WIPER LUBRICATOR

The wiper *A* takes oil off a pendent wick (see Fig. 33).

engines. Such a lubricator fitted to the cylinder of a portable engine is shown in Fig. 29. One or more deliveries are arranged as the case requires, but each delivery has its own pump plunger, the stroke of which is variable.

In the example shown in Fig. 29, the mechanical pump has adjustable feed and flushing arrangement which can be operated by hand when desired.

When mechanical lubricators are used, the small non-return or check valves in the delivery pipe need occasional attention, and the end of the delivery pipe where it enters the cylinder or steam pipe may get carbonised up and so stop the supply of oil. This should be inspected occasionally and cleaned out.

Displacement Lubricators

Engines running on saturated or wet steam are frequently fitted with displacement lubricators (Fig. 30). These, as the name implies, rely on the oil being displaced by condensed steam, the water floating the oil out of the lubricator. Sometimes a sight feed is fitted to these lubricators, each drop of oil passing to the cylinder being visible.

The method of working the lubricator is as follows. The valve connecting the lubricator with the cylinder or steam pipe is closed. A drain tap on the lubricator body is opened and the water drawn off. It is generally necessary to open the filler plug to allow the water to escape. When the water is all gone, the drain tap is closed and the lubricator body fitted. The filling plug is then replaced and the steam valve opened. If there is a sight feed, the drip is regulated by a separate valve to give any rate desired. These lubricators can be refilled while the engine is working.

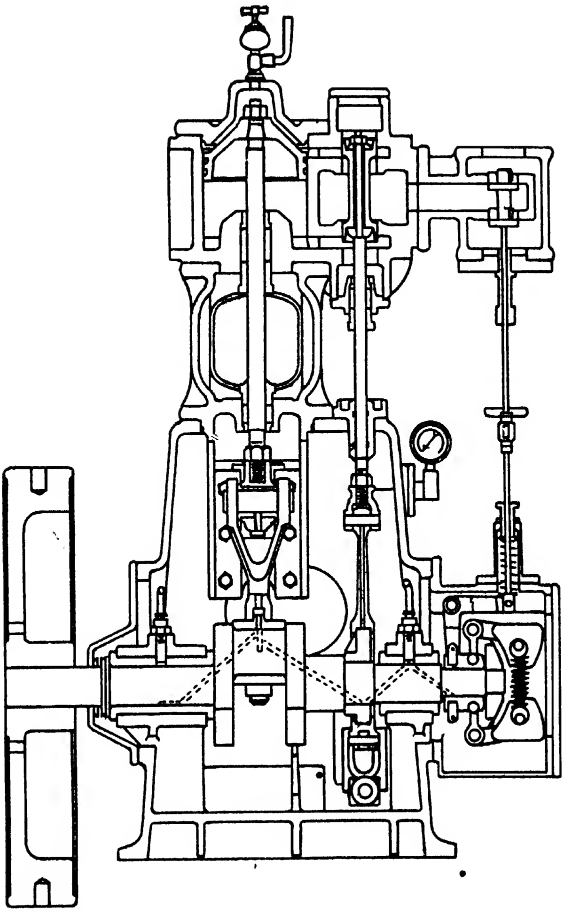


FIG. 36.—FRONT ELEVATION OF ENCLOSED HIGH-SPEED PISTON-VALVE STEAM ENGINE

Showing plunger-type lubricating pump driven off eccentric cap. Note oil passages through the crankshaft and rods (shown dotted). Also oil-pressure gauge and relief valve. The oil strainer, removable for cleaning, is shown on bottom of crankcase. (*E. Reader & Sons, Ltd.*)

Non-mechanical Lubricators

A very old type of cylinder lubricator still much used on steam winches and other engines which do not run for long periods is shown in Figs. 31 and 32. With the single cup (Fig. 31), the tap is turned off, the plug on the body removed, and the body filled with oil. The plug is then replaced and the tap turned on. The double-cup lubricator has two taps (Fig. 32). The lower is first turned off and the cup at the top filled with oil. The top cock is then opened, and the vacuum formed by the condensed steam in the body or lower cup draws the oil into the cup. The top tap is then closed and the lower opened.

Lubricating Oils for Cylinders

For low and medium pressures any good steam cylinder oil may be used, but for high pressures and superheat, special high-flashpoint oils with freedom from tarry residue are essential, and the use of an unsuitable oil on any engine using superheated steam would probably ruin the cylinder and valves in a very short time.

Oils to withstand steam at a temperature of 750° F. have for some time been available, and this temperature represents the highest in general use. On the other hand, very little damage is done on engines using very low-pressure saturated steam if the cylinder is without lubrication for quite long periods, the film of water acting as a lubricant to the piston rings and slide-valve face. One engine which worked daily for sixteen years at a pressure of 40 lb. had no lubrication of the cylinder of any sort during this period. On examination the cylinder bore, piston rings, and the slide valve and its face had a surface resembling black glass.

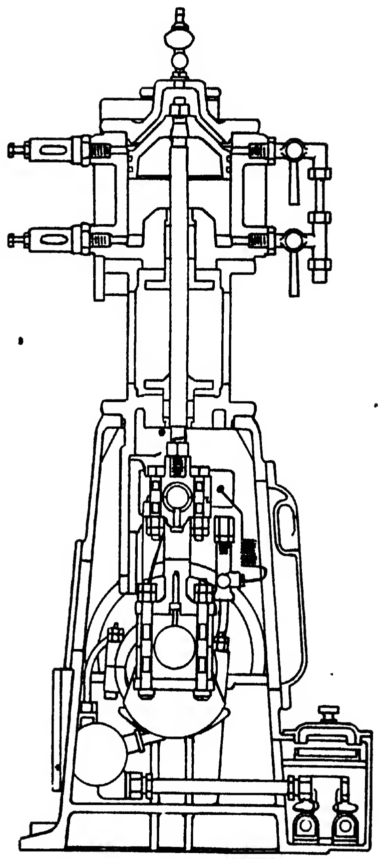


FIG. 37.—SIDE ELEVATION OF HIGH-SPEED PISTON-VALVE STEAM ENGINE SHOWN IN FIG. 36 (*E. Reader & Sons, Ltd.*)

CRANKSHAFT LUBRICATION

Ring Oiling

The bearings on crankshafts of steam engines are generally of the ring oiling type, the number of rings or chains varying with the size of the bearing. As long as the oil is maintained at the correct height in the well under the lower brass, no further attention is required for long periods, but the oil should be drained off, the well flushed out with paraffin, and fresh oil added every few months. The oil passages and channels in the brasses and block may get choked up in time, and it is advisable to inspect these periodically to see that they are clear.

Inspection lids are usually provided in the cap of each bearing, so that the attendant can readily ascertain if the rings are working properly.

In large engines oil is often circulated round the bearings by an oil pump driven off the shaft.

A four-part crankshaft bearing with ring oiling is shown in Fig. 38, whilst Fig. 39 shows a two-part bearing with ring oiling.

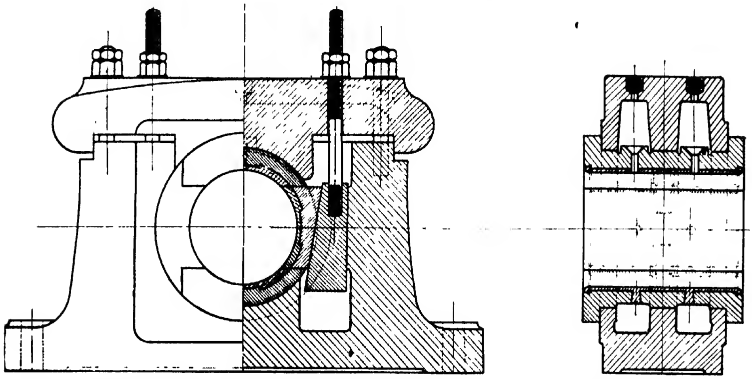


FIG. 38.—FOUR-PART CRANKSHAFT BEARING WITH RING OILING (*Robey & Co., Ltd.*)

High-speed Enclosed Engines

High-speed enclosed engines have forced lubrication in many cases. An oil pump driven by a link off the eccentric cap circulates oil through passages drilled in the crankshaft, and by holes drilled in the connecting rod the cross-head pin is lubricated. An oil filter is fitted to the pump suction and an oil-pressure gauge and adjustable relief valve enables the oil pressure to be regulated as desired. The success of this system led to its adoption for high-speed petrol and oil engines.

The oil must be maintained at a given level in the crankcase. Gauge taps or gauge glasses are fitted to give this level, and at periodical intervals the oil must be drained off, filtered, dried, and can then be reused or fresh oil can be employed.

Figs. 36 and 37 show sectional views of a high-speed piston-valve steam engine. In this engine the lubrication system incorporates duplicate strainers

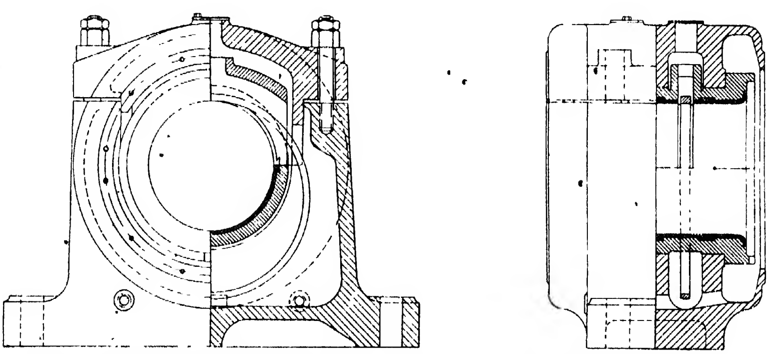


FIG. 39.—TWO-PART CRANKSHAFT BEARING WITH RING OILING (*Robey & Co., Ltd.*)

situated in an external oil box at the front of the engine. These strainers are provided with automatic flap valves which close when the strainer is removed, thus preventing unfiltered oil entering the lubrication system. As each strainer is of sufficient area to meet the needs of the engine, it is not necessary to stop the prime mover when cleaning is being carried out.

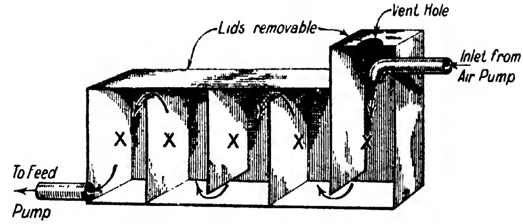


FIG. 40.—SIMPLE TYPE OF OIL SEPARATOR

Path of water is marked by arrows. The tanks *XX* are filled with coke.

Open Slow-speed Engines

On open slow-speed engines the working parts are lubricated by either sight drip feed oil cups, a wiper cup (Fig. 35) being fitted to the connecting rod and eccentric, or oil boxes with wicks (Fig. 33).

Open-type High-speed Engines

Open-type high-speed engines usually have a central oil box fitted with drip feeds controlled by needle valves or taps, the oil being piped to the various points and bearings.

When Starting Engine

Before the engine is started up all lubricators should be filled, and if of the sight-feed type (Fig. 34), the needles should be lifted for a second or so to allow a slight flush of oil to pass to the bearings. The feed should then be adjusted to give the rate of drip required.

If a mechanical lubricator (Fig. 29) is fitted to the cylinder, a few turns should be given to the lubricator by hand before starting the engine.

Testing Heat of Bearing

After the engine has been running a short time, all bearings, etc., should be felt to ascertain if there is any sign of heating, which will denote, if the bearing is properly fitted and adjusted, some failure of the lubrication. This should receive immediate attention.

Cooling a Badly Heated Bearing

In the event of a main bearing heating badly and a stoppage to attend to it being difficult, the bearing may be cooled off by using a lubricant composed of black cylinder oil mixed with mercurial ointment, $\frac{1}{4}$ lb. of ointment to 1 gallon of oil.

Another lubricant for hot bearings is composed of tallow 1 lb., flake

FIG. 41.—BADLY SCORED AND REDUCED PISTON ROD

Scoring due to neglect and undue tightening of the glands. The parts *XX* show the original size.

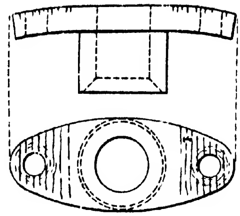


FIG. 42.—GLAND BENT BY ILL-USAGE

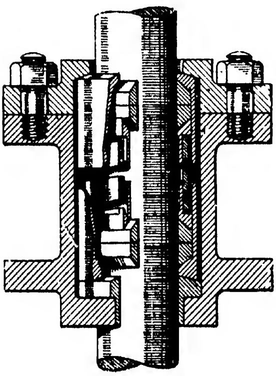


FIG. 43.—METALLIC PACKING WITH SPRING ADJUSTMENT

Note conical end bushes which are split and closed round rod by the spring pressure.

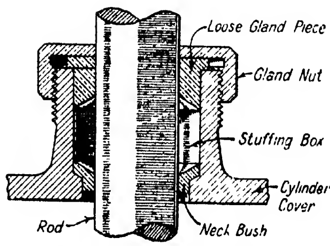


FIG. 44B.—GLAND WITH SCREWED NUT AND LOOSE GLAND PIECE

Generally used for small rods and spindles.

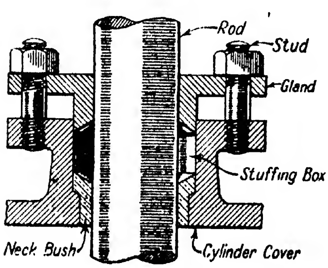


FIG. 44A.—GLAND AND STUFFING BOX —THE MOST-USED TYPE

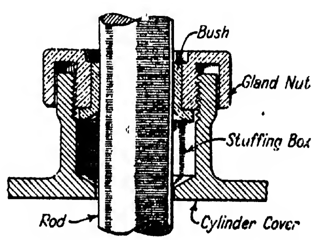


FIG. 44C.—A TYPE OF GLAND NOT MUCH USED

The internal thread tends to get choked with packing.

situated in an external oil box at the front of the engine. These strainers are provided with automatic flap valves which close when the strainer is removed, thus preventing unfiltered oil entering the lubrication system. As each strainer is of sufficient area to meet the needs of the engine, it is not necessary to stop the prime mover when cleaning is being carried out.

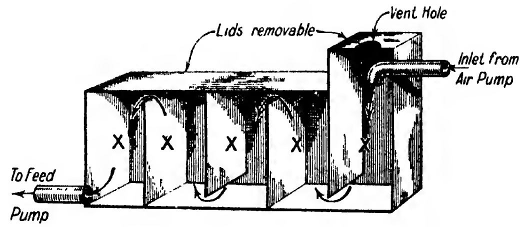


FIG. 40.—SIMPLE TYPE OF OIL SEPARATOR

Path of water is marked by arrows. The tanks *XX* are filled with coke.

Open Slow-speed Engines

On open slow-speed engines the working parts are lubricated by either sight drip feed oil cups, a wiper cup (Fig. 35) being fitted to the connecting rod and eccentric, or oil boxes with wicks (Fig. 33).

Open-type High-speed Engines

Open-type high-speed engines usually have a central oil box fitted with drip feeds controlled by needle valves or taps, the oil being piped to the various points and bearings.

When Starting Engine

Before the engine is started up all lubricators should be filled, and if of the sight-feed type (Fig. 34), the needles should be lifted for a second or so to allow a slight flush of oil to pass to the bearings. The feed should then be adjusted to give the rate of drip required.

If a mechanical lubricator (Fig. 29) is fitted to the cylinder, a few turns should be given to the lubricator by hand before starting the engine.

Testing Heat of Bearing

After the engine has been running a short time, all bearings, etc., should be felt to ascertain if there is any sign of heating, which will denote, if the bearing is properly fitted and adjusted, some failure of the lubrication. This should receive immediate attention.

Cooling a Badly Heated Bearing

In the event of a main bearing heating badly and a stoppage to attend to it being difficult, the bearing may be cooled off by using a lubricant composed of black cylinder oil mixed with mercurial ointment, ¼ lb. of ointment to 1 gallon of oil.

Another lubricant for hot bearings is composed of tallow 1 lb., flake

FIG. 41.—BADLY SCORED AND REDUCED PISTON ROD

Scoring due to neglect and undue tightening of the glands. The parts *XX* show the original size.

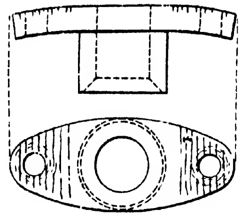


FIG. 42.—GLAND BENT BY ILL-USAGE

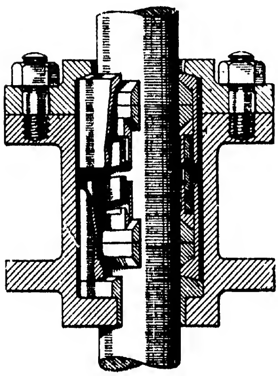


FIG. 43.—METALLIC PACKING WITH SPRING ADJUSTMENT

Note conical end bushes which are split and closed round rod by the spring pressure.

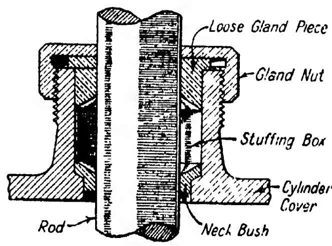


FIG. 44B.—GLAND WITH SCREWED NUT AND LOOSE GLAND PIECE

Generally used for small rods and spindles.

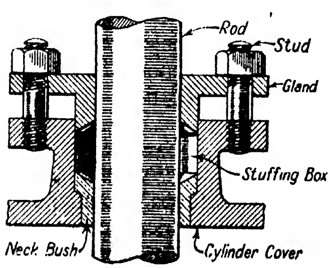


FIG. 44A.—GLAND AND STUFFING BOX —THE MOST-USED TYPE

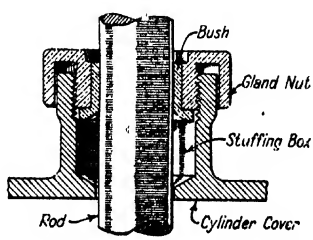


FIG. 44C.—A TYPE OF GLAND NOT MUCH USED

The internal thread tends to get choked with packing.

The old form of packing composed of canvas and rubber is seldom used now, being very severe on the rods. An unsuitable packing can cause very considerable damage, and in case of any doubt, the makers of the packing should be consulted.

If the neck bush is bevelled, a turn of round packing should be fitted first, as the square form will not readily accommodate itself to the bevel and will lead to a lot of following up being required.

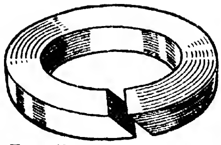


FIG. 47.—PACKING RING

Ends of rings, if cut bevelled as shown, assist entering in box.

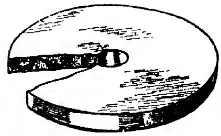


FIG. 48.—BROKEN AIR-PUMP VALVE

Moulded rubber air-pump valve broken across central hole; valve usually leaves its pin and vacuum is lost.

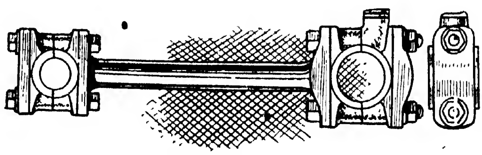


FIG. 49.—MARINE-TYPE CONNECTING ROD

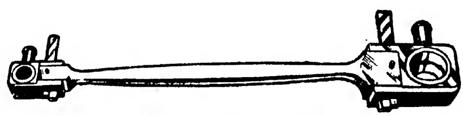


FIG. 50.—CONNECTING ROD

With both ends of the gib and cotter type.

Tightening Gland Piece

After the box is full, the gland piece should be tightened well down to "bed" the packing. It will probably be found that another turn can then be inserted. The gland piece should always be allowed to enter the box far enough to prevent it canting when tightening up.

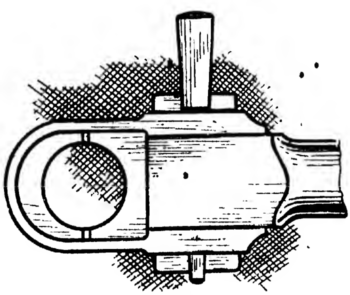


FIG. 51.—CONNECTING-ROD BEARING ADJUSTMENT

By driving in the cotter, the brasses are adjusted to take up wear

Care should be taken that the packing is not nipped between the side of the box and the gland piece when the box is full, or this will jam the gland piece against the rod and cause undue wear and damage.

Pump Glands

For the pump glands a packing termed a greasy packing is used. This is applied as in the case of steam packing, and is impregnated with a mica base lubricant. It should be well bedded, and the gland piece slightly slackened off, as most pump plungers being of gunmetal are liable to score if the gland is too tight.

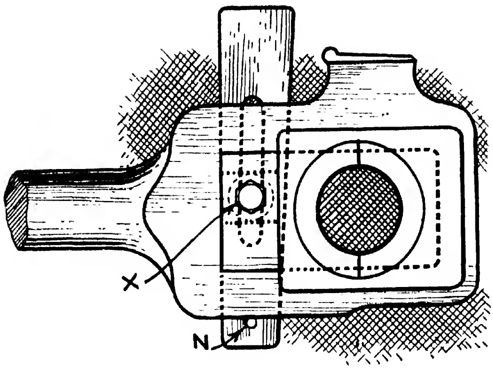


FIG. 52.—FITTING CONNECTING-ROD BEARING ADJUSTMENT

A setscrew X is generally fitted to lock the large cotter. A pin is often put through the cotter at the bottom end Z to prevent it being thrown out if it became slack.

Small Glands

For the many small glands about the engine and on the steam valves and cocks, a twist packing is used. This is composed of asbestos yarn well impregnated with graphite and oil. It can be untwisted into any number of strands to suit the gland being packed and has the advantage of never getting dry and hard.

Periodical Attention to Glands

Glands should be given periodical attention. To wait until the gland blows before tightening is bad practice, as by that time the steam may have cut a channel through the packing and only repacking will give a tight gland. Tightening should be frequent following repacking until the packing is well bedded down.

On no account should the packing be allowed to wear to such an extent that the gland piece "bottoms," as by that time there is practically no packing left in the gland and it may blow out at any time, tightening then being

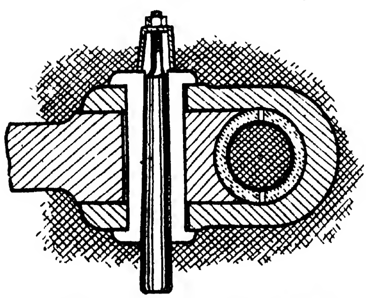


FIG. 53.—CONNECTING-ROD BEARING ADJUSTMENT

A screw adjustment is often provided to the cotter and allows of finer adjustment than Fig. 51.

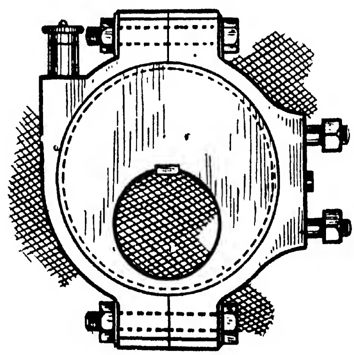


FIG. 54.—ECCENTRIC STRAP AND SHEAVE

Wear is taken up either by removing shims between the faces of the straps or by filing off the faces.

The old form of •packing composed of canvas and rubber is seldom used now, being very severe on the rods. An unsuitable packing can cause very considerable damage, and in case of any doubt, the makers of the packing should be consulted.

If the neck bush is bevelled, a turn of round packing should be fitted first, as the square form will not readily accommodate itself to the bevel and will lead to a lot of following up being required.

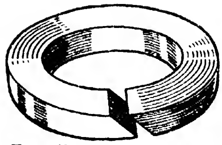


FIG. 47.—PACKING RING

Ends of rings, if cut bevelled as shown, assist entering in box.

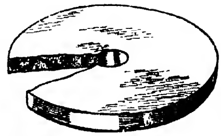


FIG. 48.—BROKEN AIR-PUMP VALVE

Moulded rubber air-pump valve broken across central hole; valve usually leaves its pin and vacuum is lost.

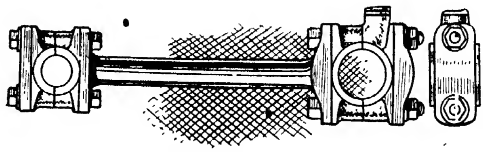


FIG. 49.—MARINE-TYPE CONNECTING ROD

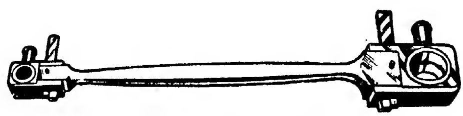


FIG. 50.—CONNECTING ROD

With both ends of the gib and cotter type.

Tightening Gland Piece

After the box is full, the gland piece should be tightened well down to "bed" the packing. It will probably be found that another turn can then be inserted. The gland piece should always be allowed to enter the box far enough to prevent it canting when tightening up.

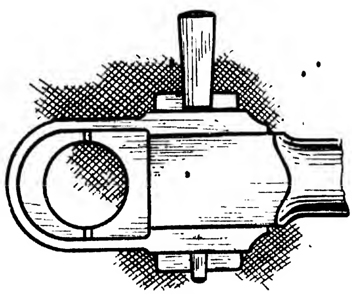


FIG. 51.—CONNECTING-ROD BEARING ADJUSTMENT

By driving in the cotter, the brasses are adjusted to take up wear

Care should be taken that the packing is not nipped between the side of the box and the gland piece when the box is full, or this will jam the gland piece against the rod and cause undue wear and damage.

Pump Glands

For the pump glands a packing termed a greasy packing is used. This is applied as in the case of steam packing, and is impregnated with a mica base lubricant. It should be well bedded, and the gland piece slightly slackened off, as most pump plungers being of gunmetal are liable to score if the gland is too tight.

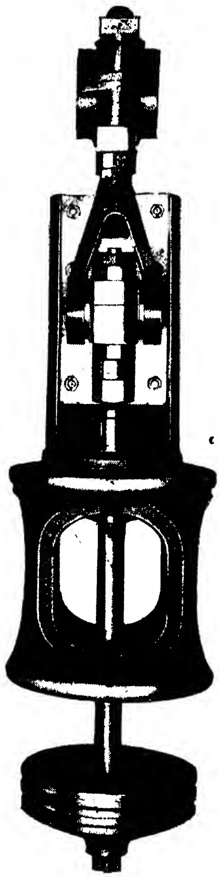


FIG. 57.—PISTON-ROD LINE AND CROSS-HEAD ASSEMBLY

Used on high-speed enclosed engines. See Figs. 36 and 37. (*E. Reader & Sons, Ltd.*)

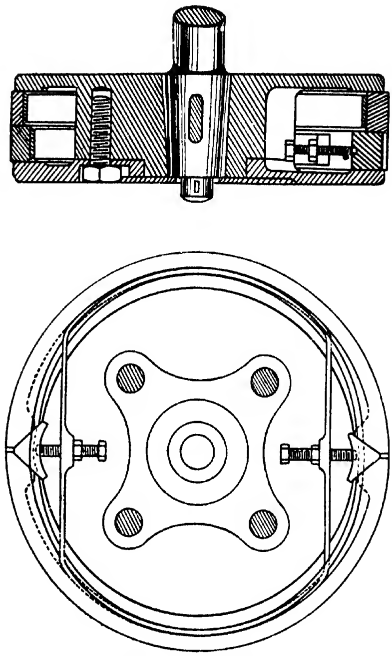


FIG. 58.—LARGE JUNK-RING TYPE PISTONS

Note taper blocks carried on a spring frame with setscrew adjustment for opening rings to take up wear. Only one ring shown in plan, the top ring and its spring frame being removed.

Many flange and steam chest joints are made with ribbed metal gaskets, usually of annealed copper, though brass is sometimes used. It is advisable with these joints to use a liquid or plastic jointing material to ensure tightness, and the joints must be followed up when hot.

Attention to Air-pump Valves

In condensing engines, the air-pump valves must receive periodical attention. These are generally disc valves of moulded rubber composition, and in time tend to break through as shown in Fig. 48. The failure of a valve will seriously affect the vacuum, and a drop in the vacuum is usually a sign of valve trouble if the circulating water is ample and of the usual temperature. Some pumps are fitted with metal valves which are less likely to fail and are only supplied by the makers of the pump.

Bearing Adjustments

The crankshaft bearings on large steam engines are generally of the four-piece type. Practically all other steam engines are fitted with the two-piece type. The centre blocks are adjusted up by screw wedges acting on the back of the blocks. The wedge screws with their locknuts can be seen on the top of the pedestal. This enables a very fine adjustment to be made, and in practice a very long period of running is possible before the brasses need any attention or renewing. The pedestals in the larger sizes are cast solid with the engine-bed plate.

The ordinary types of crankshaft bearing with two-piece brasses split either vertically or horizontally are shown in Figs. 38 and 39. The brasses in the latter are adjusted by setscrews in the side of the pedestals. The horizontally split brasses are set together in the usual way by filing or milling off the edges and then scraping the brass to the shaft.

Connecting Rods

The connecting rods used on steam engines are of two general types, the marine (Fig. 49) and the gib and cotter (Fig. 50). Some rods have a combination of both, the big end being of the marine type with the small gib end and cotter, and vice versa.

Taking up Wear in Gib and Cotter Ends

A gib and cotter big end is shown in Fig. 51, and it will be seen that by driving in the cotter the inner brass is forced forward, thus taking up the wear, a sufficient gap being allowed in the brasses to enable considerable adjustment to be made. The large cotter is then generally locked by a setscrew in the side of the rod (Fig. 52). In many cases a screw adjustment is fitted to this cotter (Fig. 53), which enables a much finer adjustment to be made, and is preferable to Fig. 51.

Taking up Wear in Marine-type Ends

In the marine-type ends, shims, or thin pieces of packing either of steel or brass, are fitted between the cup and the road, and adjustment is made by taking out one or more of these shims as required.

Eccentric Straps

The eccentric straps (Fig. 54) are also frequently fitted with shims, and can be adjusted by removing these, but an eccentric strap should always be on the slack side. The bearing surface is very great and the least suspicion of tightness will cause rapid heating up, and should the strap grip up on the shim, the rod will be broken, with disastrous results.

Taking up Wear in Eccentric Rod Pin

The pin connecting the eccentric rod to the valve spindle guide (*C*, Fig. 55) is subject to a heavy load, especially in the case of slide-valve engines. To take

up wear in the pin, a device as shown in Fig. 55 is often fitted. By slackening the setscrew and driving in the cotter, the bronze block is tightened on the end of the eccentric rod and takes up any wear in the pin. The same method is often used to take up the wear in the pump plunger as in Fig. 21.

Crossheads

There are many different types of motion blocks or crossheads, some adjustable, others not. Actually the lack of adjustment is not a serious matter, as the life of a crosshead, with the slipper blocks, is very long, and many years' work is required to produce any appreciable wear. The crossheads of nearly all engines are of the bored guide type.

Pistons and Rings

In high-speed engines, rings and pistons of the Ramsbottom type are generally used. Such a piston is shown in Fig. 57.

The rings, which are of steel, when worn are simply sprung out of their grooves and new rings sprung in.

If the grooves are worn, these are trued up in the lathe and new rings of a wider section fitted.

The ends of the rings, when sprung into the size of the cylinder bore, should not butt, but have a space of 0·001 in. per inch of diameter to allow for expansion.

SPRING RINGS—ROWAN TYPE.—There are many patent types of spring rings used in larger engines. One, the Rowan, is shown in Fig. 56, for a solid piston, but is also supplied for junk-ring pistons. By the time such rings need renewal, the cylinder will need reboring, so it is a question of repair rather than adjustment. Large engines generally have junk-ring type pistons, as shown in Fig. 58. These are adjustable, and the illustration is self-explanatory.

Governors and Governor Gear

The chief wear in governors and governor gear is in the pins and links, and in many cases no provision is made for adjustment. When wear becomes pronounced, new pins or bushes are fitted. Fig. 19 shows a well-known type of expansion governor, and the number of small pins and links will be noted. Although these are generally hardened and ground, wear takes place after long service, and renewal of the worn parts is the only satisfactory remedy.

STEAM-ENGINE TESTING

THERE are two distinct methods of testing steam engines:

The first type of test is known as the brake horse-power method. This is a purely mechanical test for determining what power is delivered by the engine to the crankshaft. The methods of conducting this test are, in the main, similar to those described in the section beginning on page 71.

The second type of test is known as the indicated horse-power test, which determines the actual power developed in each of the engine cylinders.

The Object of the Indicator

The object of the steam-engine indicator is to provide a diagram showing the steam pressure in the engine cylinder at all positions of the piston. This diagram is of great importance to the engineer, as it gives a complete record on paper of the events taking place in the cylinder, and it can therefore be used as a means for ensuring economy in running. Some of the uses to which the indicator diagram is put are given on page 434. First, however, let us consider the instrument itself.

Medium-speed Engine Indicator

Steam-engine indicators vary in design according to their make and type, but for the purposes of this article it is proposed to consider the medium-speed engine indicator shown in Fig. 1.

It consists of a piston and cylinder connected by means of a

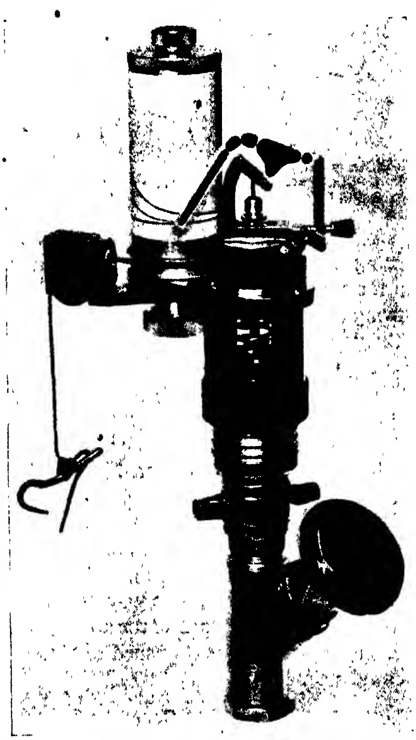


FIG. 1.—DOBBIE-McINNES MEDIUM-SPEED ENGINE INDICATOR

This may be used both for steam and diesel engines. For steam use a cock can be substituted for the diesel valve shown.

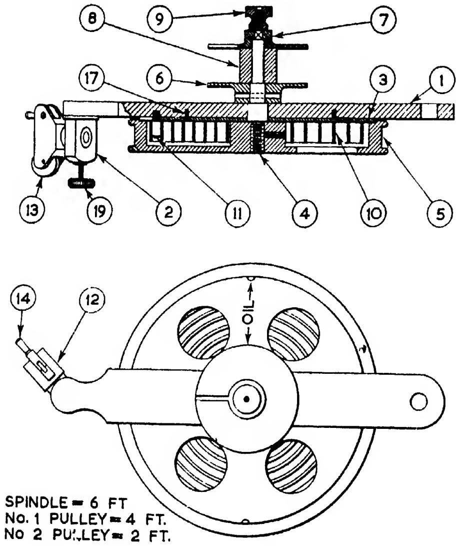


FIG. 2.—DOBBIE-MCINNES "WADE" REDUCING GEAR

1. Bracket.
2. Pulley bracket.
3. Spring-case cover.
4. Spindle.
5. Large pulley.
6. Lower flange.
7. Upper flange.
8. Pulley.
9. Thumbnut.
10. Spring.
11. Spring stop.
12. Pulley block.
13. Pulley wheel.
14. Cord detent.
17. Securing screw.
19. Pulley thumbnut.

cock to a tapped hole in the engine-cylinder wall. An air-cooled calibrated spring is fitted to the piston rod, so that the travel of the rod is proportional to the steam pressure acting on the piston, and this spring is made specially strong to reduce the piston travel, thus minimising the inertia effect which would otherwise tend to distort the diagram.

To produce a diagram of readable proportions, the piston rod is fitted with a parallel motion, or link gear, working on the pantograph principle, and reproducing exactly the movements of the piston but on a larger scale. One link carries a pencil with which the diagram is drawn.

The diagram paper is attached by clips to a reciprocating drum, which is connected through a stroke-reducing gear—referred to later—to the crosshead of the engine. In this way, every movement of the engine piston is reproduced exactly by an equivalent movement of the indicator drum, but on a small scale. Diagrams may be up to 2½ in. high.

The indicator cock is two-way, so that when the indicator is shut to the engine it is automatically opened to the atmosphere; also, when the indicator is isolated, the engine or tail pipe is opened to atmosphere to allow for clearing away condensed steam.

The Indicator Reducing Gear

Owing to the importance of using an accurate reducing gear to minimise errors of the diagram in a horizontal direction, illustrations of correct (Figs. 2–4) and incorrect gears (Figs. 5 and 6) are shown. Fig. 2 shows the Dobbie-McInnes "Wade" gear, which is of the differential pulley type. The gear is

bracketed to the indicator under the recording drum, and by fixing upper pulleys of the requisite diameter it can be made suitable for any stroke from 6 ft. downwards. The cord guide on the gear can be set at any angle, and has a clutch or detent brake to stop motion of the drum, thus allowing the diagram paper to be changed.

The lever-type gear (Fig. 3) is frequently used on marine engines, and is fitted by the engine builder. Those who know the properties of "similar triangles" can verify that this gear is exact, and that the movement of the indicator lead is proportional to that of the crosshead. Another gear having similar properties is shown in Fig. 4.

Points which apply to all Types of Gear

It is difficult to name any one type of gear that can be used universally, as many factors, such as space available, position of indicator relative to crosshead, engine speed, etc., require consideration. The following points should be observed, however, for all types of gear:

1. The stroke reduction should be such that the indicator drum rotates backwards and forwards without reaching either of its two stops.
2. The motion of the indicator lead should always be proportional to that of the engine piston.
3. There must be no play or slackness in the gear.
4. The indicator lead should be taut, as inelastic as possible, and free from kinks.
5. Change-direction pulleys for the lead should be few in number, light, of comparatively large diameter, well lubricated, and not slack.

Preparation for Test

Suppose indicator diagrams are required from a triple-expansion marine steam engine. As there are three cylinders, three indicators and three reducing gears are required.

A reducing gear is therefore fitted to each crosshead, and a copper pipe is led from the top and bottom of each cylinder to a change-over cock screwed to take the indicator cock. This enables a diagram to be taken of the steam pressures above and below each of the three pistons of the engine without using six indicators, and the diagram from the top of each cylinder is taken on the same card as that from the bottom and immediately after it.

On screwing the instruments in position and adjusting their drumcord pulleys, the indicator cord, which is specially made for the purpose, is stretched from the reducing gears to the drum cords and suitable lengths are chosen so that each diagram shall be positioned midway along the diagram paper.

The indicator pistons are removed, cleaned, and oiled, and each is fitted with a spring of such strength that the particular maximum pressure will raise the pencil to a point on the diagram paper just below the top stop; this ensures the largest possible diagram being obtained. On reassembling the pistons, the indicators are ready for test.

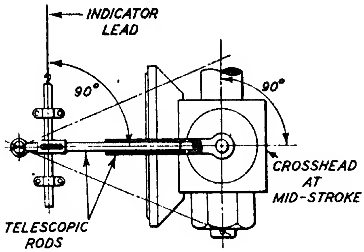


FIG. 3.—LEVER-TYPE REDUCING GEAR

The above type is frequently used in marine practice and is fitted by the engine builder. The movement of the indicator lead is proportional to that of the cross-head, this type of gear being quite accurate.

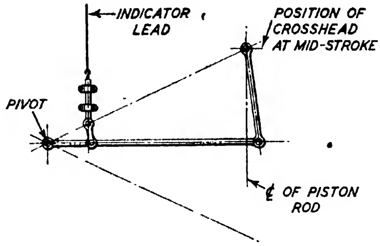


FIG. 4.—CORRECT REDUCING GEAR OF LEVER TYPE

Gear having similar properties to that shown in Fig. 3. Movement of lead is proportional to movement of engine piston. Note the "similar triangles" made by position of the different levers.

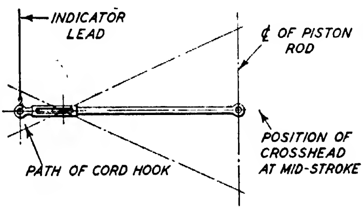


FIG. 5.—FAULTY REDUCING GEAR OF LEVER TYPE

This type of gear does not give an accurate diagram. The movement of the lead is not proportional to the movement of the engine piston. The path of the cord hook is curved. The use of such a gear should be avoided.

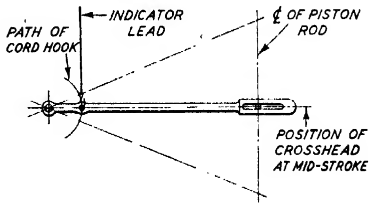


FIG. 6.—ANOTHER FAULTY REDUCING GEAR OF THE LEVER TYPE

Here again the movement of the lead is not proportional to the movement of the engine piston. Note that the cord hook takes a curved path. It is very important that an accurate reducing gear be used in order to minimise errors of the diagram.

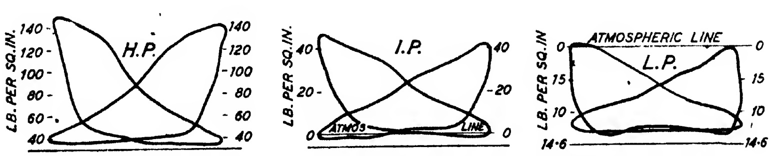


FIG. 7.—SET OF DIAGRAMS FROM THREE CYLINDERS OF TRIPLE-EXPANSION MARINE ENGINE

The width of the diagrams represents a piston stroke of 39 in., and the strengths of the springs used are indicated by the scales of pressure shown alongside each card, the atmospheric lines representing zero. The atmospheric line is not shown on the high-pressure card, but its position is easily found from the pressure scale.

It might be mentioned here that the strengths of indicator springs are engraved on their heads, i.e. a spring marked with a strength of 40 means that a pressure of 40 lb. per square inch on the indicator piston will raise the pencil 1 in.

For a condensing engine, or for the low-pressure cylinder of a triple-expansion engine, part of the diagram is below the atmospheric line, and a special light spring is used to allow for this. The markings of such a spring may be, say, "10," that is: 10 lb. per square inch for every inch of pencil travel, and "V+ 7," that is: it may be used from a vacuum or from pressures below atmospheric up to a maximum of 7 lb. per square inch above atmospheric.

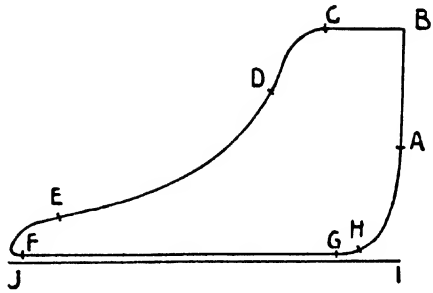


FIG. 8.—TYPICAL STEAM-ENGINE DIAGRAM

AB, admission line. *BC*, steam line. *D*, point of cut-off. *DE*, expansion line. *E*, point of release. *FG*, exhaust or back-pressure line. *H*, point of compression. *HA*, compression line. *IJ*, atmospheric line.

Taking the Diagram

By means of the indicator cock, the instrument is opened to atmosphere and the pencil is placed against the diagram paper. The drum cord is then pulled by hand to draw the atmospheric line from which pressure measurements are made.

The cock handle is turned so that condensed steam can be cleared from the tail pipe, which is afterwards put into communication with the indicator, the pencil meantime being removed from the drum while the cord is hooked to the lead from the gear. A short time is allowed for the steam to warm the indicator, and a diagram is then obtained.

When a branch pipe is used, the process is repeated for the other end of the cylinder, thus obtaining a second diagram on the same card, the atmospheric line being common to both outlines.

Fig. 8 illustrates the essential features of a steam-engine diagram. Steam is admitted to the cylinder at *A* and the pressure rises to *B*, the piston being on top dead centre. This causes the piston to move to *C*, steam still entering. At *C*, the slide valve begins to close, the steam supply being completely cut off at *D*. From *D*, the steam expands in the cylinder, driving the piston before it, until at *E* the slide valve opens and allows the steam to escape to exhaust. The piston reaches the bottom centre, and during the return stroke pushes the remainder of the spent steam through the exhaust port, until at *H* the slide valve has closed. From *H* to *A*, what steam is left in the cylinder is compressed to the pressure at *A*, when the cycle is repeated.

Uses of the Diagram

The *area* of the diagram which represents the work done during the cycle is used for finding the indicated horse-power (I.H.P.) of the engine, that is, the power delivered by the steam in driving the engine itself and in propelling the ship or the machines to which the engine may be coupled.

The *outline* of the diagram, showing the positions of admission and release, the amount of initial, maximum, and back pressure, and the extent of compression and expansion, gives a good indication of correct or incorrect valve setting which, if incorrect, may seriously increase consumption of steam and fuel without addition to the power output.

Two other valuable calculations, using the indicator diagram, are made to find the efficiency of the engine: one is the determination of the hourly consumption of steam for every I.H.P. developed, and the other that of the mechanical efficiency (B.H.P. ÷ I.H.P.) which shows how much energy is necessary to overcome friction in the engine.

THE OPERATION AND MAINTENANCE OF STEAM TURBINES

SUCCESSFUL operation of steam turbines depends to a great extent on the care and attention given to the machines in service, by which enforced "outages" for maintenance are kept to a miminum.

Steam turbines operating under "ideal" conditions generally require the least amount of maintenance, but unfortunately "ideal" conditions are seldom met with in practice. The nearest approach to such conditions are met with on "base load" machines, operating at moderate thermal conditions. Such machines are known to give years of satisfactory service without requiring major repairs.

Effects of Faulty Operation

To illustrate the importance of good operation, some of the extreme effects of faulty operation are mentioned below.

NON-UNIFORM COOLING.—Both the steam-turbine casing and shaft may "bow" because of non-uniform cooling after unloading, and during the subsequent start the shaft may rub the gland and labyrinth packings and bow still farther. The resulting unbalance can increase to such an extent that severe damage may be caused. Turbine makers are well aware of this possibility, and where there is the likelihood of such an occurrence they generally equip their machines with "turning" or "barring" gear, which enables the shafts to be turned at low speeds during shut-down cooling periods and subsequent "warming-up" periods in preparation for starting. Operators should be well versed in the intelligent use of "turning" gear.

CRUSHING AND VIBRATION.—Since the rate of heat transfer differs greatly in the various parts of a machine, crushing or distortion of internal fits and surfaces may occur, and the resulting stresses may permanently distort the parts.

When rotating and stationary parts expand at different rates, mechanical contact may occur, with consequent wear on the rubbing strips which are intended to minimise steam leakage between stages. The efficiency of the machine would fall off, but no serious mechanical damage would be expected, unless such rubbing set up vibration of the rotating parts, when the effect could be as severe as that caused by an unbalanced shaft due to "bowing."

DISTORTION BY WATER "SLUGS."—An infrequent but unusually severe condition occurs during emergencies, when water "slugs" are carried over from the boilers. Thrust-bearing failures, blade failures, and permanent distortion may result.

It is known that diaphragms will distort when heated or cooled rapidly, and this may result in small permanent distortions which become cumulative and measurable after many cycles.

Enough, then, of some of the effects of faulty operation; let us rather consider what should be good operation of steam turbines.

DESCRIPTION OF A MEDIUM-SIZED STEAM TURBINE

Machines of various makes, types, and sizes differ so much that it would be impossible in an article of this length to more than outline them. Let us therefore consider a "composite" machine of what to-day would be termed "medium size."

Construction Details

Such a machine would be a 30,000-kW. M.C.R. output, at 3,000 r.p.m., with steam input at 600 p.s.i.g. and 850° F., exhausting to twin condensers maintaining a vacuum of 28–29 in. Hg. It would be of 2-cylinder design, the H.P. cylinder being of the impulse type and the L.P. of the double-flow reaction type, i.e. the outlet from the H.P. cylinder would be led to the middle of the L.P. cylinder, the steam path then being divided to fore and aft to twin condensers, one to each exhaust opening.

Speed of Shafts

With the need for keeping gland diameters as small as possible, the H.P. shaft would have its critical speed in the order of 30–60 per cent. of the running speed. The L.P. shaft of reaction type could be of shaft or drum construction, and would be considered to have its critical speed above running speed. Modern investigations indicate that belief in stiff shafts is not to be relied on too implicitly, as the modifying effect of bearing supports may so lower the critical speed as to bring it close on the running speed, where the effect would be worse than if the critical speed were designed to be well below the running speed.

Inlet and Outlet Casings

The H.P. inlet casing would be of carbon-molybdenum cast steel, with the H.P. outlet casing of carbon cast steel. The L.P. inlet casing would be of high-grade cast iron, while the exhaust casings would be of fabricated mild steel.

The steam connection between H.P. and L.P. cylinders would be through welded steel pipes provided with expansion pieces.

The shafts would be of nickel steel.

Blading Design

The blading of the earlier stages of the H.P. cylinder would be of HECLA A.T.V. steel, while the remaining stages and those of the L.P. cylinder would be of stainless steel. Where there is the impossibility of keeping blades of certain dimensions free from harmful vibrations at the running speed, the manufac-

turers will have resorted to "lacing" or stiffening certain stages. Prevention of blade vibrations is almost entirely in the hands of the designer, but his best efforts are nullified when a machine is operated at other than its designed speed, either above or below, and operators should always bear this fact in mind.

The Governor

Throttle or nozzle governing would be adopted, the first generally for a reaction-type machine, and either method for an impulse-type.

The governor would be of the sensitive centrifugal type, totally enclosed, operating through worm gearing at a speed of from 15 per cent. to 30 per cent. of the turbine speed. The governor and gear should be capable of maintaining the machine speed constant, and have "drooping" characteristics, i.e. a slight fall in speed with increasing load, the fall being not more than 4 per cent. from no load to full load. "Speeder" gear would be provided to maintain constant speed at any load, and this speeder gear should be suitable for remote control. The governor should also be capable of preventing a momentary speed rise on the sudden loss of full load of not more than 10 per cent. above normal speed.

One or two emergency governors of the eccentric ring type would be fitted, which through associated gear would shut off steam should the machine for any reason exceed 10 per cent. above normal. These are sometimes termed "runaway" governors.

The above brief description will be our typical machine on which to consider operation generally, and which will be dealt with under the following headings: Starting and Stopping, including Heating and Draining; Glands and Packings; Lubrication; Governors and Emergency Governors.

STARTING AND STOPPING, INCLUDING HEATING AND DRAINING.

When starting up from cold, the admission of steam to the turbine is accompanied by condensation of some of the steam, as the casing structure and internal parts of the machine are much lower in temperature than the admitted steam.

Provision for Drainage

This condensed water must be removed from the path of rotating blades, and also from such parts that normally operate where the steam is superheated.

Ample provision for drainage is always provided. Some points, such as stop valves and steam chests, drain externally, and are normally connected to steam traps fitted with bypassing arrangements. There has been a tendency in recent times to cut down these external drainage points, it being assumed that with superheated steam there will be no drainage except at starting times. Many operators have found to their cost that even in the best-regulated steam-raising plant, water does find its way into the turbine unless care is exercised.

Most other points inside the turbine drain internally to the condenser, though there are some points towards the high-pressure end which need draining only when starting or on light loads. These are usually under the control of the operator, and may be drained externally. These latter drains, though necessary during light loads, are not necessary during medium and high loads. If left open at those periods the drains will prove a source of considerable loss.

Working Temperature

To-day it is recognised that all "warming up" must be done gradually and with rotating parts turning, so that they attain their working temperatures as uniformly as possible.

A machine such as we are considering, and with inlet steam temperature of 800° F. or above, would be fitted with motor-driven "turning gear" to rotate the shafts at from 3 r.p.m. to 60 r.p.m. during periods of starting and stopping. The turning gear should be set in motion, and as the machine is quite cold the duration of "turning" need only be short, a matter of 15 minutes or so. Of course, the necessary auxiliary oil pumps would have first been started.

The Turning Gear

During this period, while the turning gear is in operation, the operator would start up the remainder of the auxiliary plant, such as circulating water-pumps, extraction condensate pumps, and air-extraction plant for the condenser.

Cooling water should be shut off from the oil coolers, to enable the oil to reach its operating temperature as quickly as possible. During this period also, a rapid survey should be made of the oil circulation, and of the machine in general to see that there is nothing amiss.

Everything being normal, the gland-sealing steam should now be turned on and regulated so that a "wisp" of steam issues from the gland vent pipes. Partial vacuum of from 15 in. Hg to 20 in. Hg should be established, when live steam may be admitted via the throttle or control valves sufficiently to rotate the machine at about 10 per cent. of its normal speed. When the speed of the rotors under steam admission exceeds that of the "turning" or "barring" speed, the turning gear should disengage, though some makers recommend that the turning gear be first disengaged and then live steam admitted to the machine.

Steam Control

At this stage, admission of live steam to the machine would be under the control of the operator, who would also attend to the drainage as the machine warmed up and the condenser vacuum became fully established.

Steady Speed Maintenance

Care should be exercised in maintaining a steady speed increase on the machine, and the instructions, as issued by the makers, for bringing a "cold" machine into service faithfully carried out. Such instructions allow ample time for temperature changes on the machine to be gradual and not to exceed approximately 200° F. per hour.

Operating Temperature

Present-day ideas for starting, stopping, or applying load on machines are based on this gradual temperature change of the metal masses, that is to keep temperature gradients as low and as uniform as possible. Thus a cold machine takes longer to warm up and reach operating condition than a "hot" machine, i.e. one recently having been on load. Likewise, the nearer a machine is to operating temperature, the shorter should be the time of run up and loading, as it is now realised that a long run-up time actually cools down a machine, whereas the effort should be to bring the machine quickly to that load corresponding to its mean temperature, the whole aim being to minimise the temperature stresses on the turbine mass.

Loading Rates

For the machine which we are considering, the following table may be used, and the loading rates for other sizes of machines interpreted as percentage load:

STARTING AND LOADING TIMES

Mean Turbine Temp. ° F. H.P. End	*Run-up Time. Minutes*	*Loading Rate. Mw./Min.*	*Loading Rate. Percentage Load/Min.*
150	40	1·0	3
350	20	1·2	4
500	10	1·5	5

An initial steam temperature change of 200° F. per hour may be taken as equal to 1 megawatt per minute load change.

With the turbine at any "steady state" load, above 20 per cent. load, i.e. after attaining steady temperatures corresponding to the load, the load can be increased to any other value at the rate of 20 per cent. full load increase per minute at constant throttle steam pressure and temperature.

At any steady load, the steam temperature may be increased at the rate of 400° F. per hour, but it must not be allowed to exceed the manufacturer's stated maximum temperature.

The above may not be possible on some types of machines, and the operator must keep to the limits set by the makers.

Unloading

Unloading, or decrease of throttle steam temperature, must be carefully considered, since the H.P. rotor contracts faster than the casing, and may result in loss of axial clearance on some machines. Here again the maker's instructions must be followed. The cooling gradients are usually low and uniform, and provided that the initial steam temperature is maintained, unloading at the rate of

20 per cent. full load per minute is possible. Likewise, so long as the steam is superheated, the steam temperature may be dropped at the rate of 400° F. per hour at any steady load.

The safest way to unload a machine is to drop all load instantly, provided the machine is shut down and the turning gear put into operation. Such a condition of unloading is rarely possible, due to the upset on the remainder of the system.

Prevention of Vibration at Critical Speeds

Reverting now to the machine being considered, the operator should carefully watch the machine as the speed is increased, and should any roughness or vibration set in, particularly at the critical speeds of the shafts, no further acceleration must be attempted; but the machine speed should be lowered to about 10 per cent. of full speed, and further time be allowed to warm up before again trying to bring up to speed. As mentioned at the beginning of this article, attempts to rush a machine through periods of vibration are foolhardy, and can only result in damage.

Viscosity of Lubricating Oils

Turbine lubricating oils have their correct viscosity in use at an inlet oil temperature to bearings of 100–110° F. Below these temperatures they are not fluid enough for high-speed bearings, so until a temperature of about 85° F. is reached, the machine speed should not exceed approximately 75 per cent. of its normal running speed.

The machine should be brought up to speed in accordance with the Table of Starting and Loading Times and then put on load.

Shutting Down the Turbine

First, the load is removed, and then the steam supply is shut off so that the machine coasts to rest. A number of operators record the running-down time of their machines. This test, of course, should be under identical conditions as regards vacuum and oil temperature.

See that the lubricating-oil supplies are maintained, and also the seals to the glands. For the type of machine under consideration the "turning gear" should be put into operation immediately the machine comes to rest, and should be kept in operation until such time as the machine is required for further load, if this time does not exceed twelve hours. If the machine is not required for a period exceeding twelve hours, the rotors should be turned one half-turn every hour of the next twelve hours and every two hours for the next twenty-four hours, or until the turbine mass has cooled to approximately room temperature.

The Function of the Turning Gear

The turning gear serves two purposes: (*a*) to maintain the shaft in a normal or unbowed condition, and (*b*) to carry away from the bearings such heat as would flow along the hot shaft; the bearings being in the meantime supplied

with cool oil while turning. The condition (*b*) would be satisfied with the turning gear in use for a period of about six hours, but condition (*a*) is only satisfied when the turbine mass approaches room temperature, and which normally with a well-lagged machine takes approximately fifty hours. An alternative method to the half-turn at regular intervals to satisfy condition (*a*) would be to use the turning gear continuously for a period of about four hours before starting up a partially cooled machine.

GLANDS AND PACKINGS

Turbine glands may be considered under two headings: (*a*) external glands, and (*b*) internal glands. The purpose of (*a*) is to minimise the leakage of steam at the high-pressure end outwards, and on the low-pressure end to prevent the ingress of air. The purpose of (*b*) is to prevent steam leakage from stage to stage inside the machine, and generally applies to impulse-type turbines.

Construction

External glands may be of the labyrinth type, carbon rings, or water seal. The carbon-ring type are somewhat limited in size, depending on the rubbing speed between shaft and packing, so that for the size of machine under consideration, the packings would probably be a combination of labyrinth and water seals.

Operation

As external glands serve two purposes, as explained above, they are usually operated in conjunction with each other, that is, the outward leakage at the high-pressure end is used to seal the low-pressure end. The various manufacturers have different methods, generally alike in principle, to attain this end, which only become operative at above a certain load, and depend on the condition of the glands.

Regulating Valves

For our typical machine, the piped connections to the glands should have regulating valves to each gland; and would need "live" steam supplies to seal under "starting up" and "no load" conditions. As the "leak off" from high-pressure glands would vary, depending on the load carried, a "regulator" would be provided to automatically adjust the steam supply to the glands, the excess passing through the regulator to the lower stages in the machine.

Other refinements would probably be present, depending on the turbine manufacturer's requirements.

Glands are necessary evils, and at best account for some percentage of the turbine steam consumption. One can readily understand the loss of efficiency that may arise from poorly maintained glands and packings, and this should not be tolerated in a well-run power-station.

Internal glands are self-sealing, in that they require no steam supply for

starting purposes. They are the best expedient available for use in the limited space between stages, but the same remarks apply as regards losses and maintenance.

LUBRICATION, INCLUDING BEARINGS AND THRUST BEARINGS

Steam shafts when rotating are supported on a wedge-shaped oil film formed between the spinning journal and the bearing white-metal lining, so that the shaft does not come into metallic contact with the bearing lining. This oil film is formed mostly by the speed of rotation of the shaft and not by the pressure of the oil supply.

The Functions of Lubrication

Without this oil film, the heat generated by friction between the shaft and the bearing would be so great as to cause "seizure," with its resultant damage.

Lubrication, therefore, serves a twofold purpose: (*a*) to minimise friction by supporting the shafts on an oil film, and (*b*) to cool the shaft journals which become hot due to conduction along the shaft from the steam spaces.

It is therefore essential to maintain an ample supply of cool, clean, and correct-grade oil to each bearing and thrust bearing.

Estimating Oil Requirements

Considering again a machine of about 30 mW. size, a reserve of oil of approximately 2,000 galls. would be required, this being pumped to each bearing at a pressure of from 10 to 20 p.s.i. Each bearing (depending on its size) would require about 10–20 galls. per minute, and thrust bearings about three times as much. The bulk of the oil would be in a "settling tank," where impurities picked up in circulation would have a brief time to settle.

Pumping the Oil

Manufacturers have various means of pumping the oil. Some have impeller-type pumps on the turbine shaft, others have "positive" displacement pumps of the meshing-gear type, also driven from the turbine shaft; but such pumps are lower in speed than impeller type, and are therefore driven through gearing.

Positive displacement pumps generally deliver at high pressure (up to 100 p.s.i.) through oil coolers and also provide power for governing relays, etc.; finally the oil at a lower pressure passes via relief valves to the bearing supply line.

Turbine Oils

Suitable oils for turbine use are as quoted in B.S. 489 : 1933, with which every operator should be familiar.

It is hardly necessary to state that the oil must be clean, as any foreign substance, particularly of a "gritty" nature, would rapidly cause damage. Rust as a foreign substance should particularly be excluded, as it is a powerful catalyst, causing "break down" of lubricating oil.

Oil supply companies everywhere may be consulted regarding suitable turbine oils, and they will no doubt stress the danger of mixing oils of various grades.

High-pressure Lubrication

As an oil film is only established at a certain velocity (generally a few feet per minute), it is sometimes necessary on very large machines and those fitted with turning gear to provide a further oil supply to the bearings at extra high pressure (up to 1,000 p.s.i.) to float the shafts at low speeds. This supply is led to the bottom of the bearings, while the normal lubricating supply at 10 to 20 p.s.i. enters the bearings at about the shaft axis.

Thrust bearings, whether of the multi-collar, Michell, or Kingsbury type, require as careful or even better treatment than journal bearings.

GOVERNORS AND EMERGENCY GOVERNORS

The governors of steam turbines follow the same general pattern, but space does not permit detailed description of the construction by various makers.

The governors normally operate at about 15 per cent. of the turbine speed, driven by the turbine shaft through reduction gear of the worm-and-wheel type.

Speed Control

The Specification for Steam Turbines, B.S. 132: 1930, states that the permanent speed variation in a governor must not exceed 4 per cent., i.e. the change in speed from no load to full load must not exceed 4 per cent. of the rated speed, and that the governed speed must be capable of adjustment within $\pm$ 5 per cent. of the rated speed. This adjustment is usually brought about by a spring loading additional to the governor itself, this spring loading being adjustable by either direct or remote control. Further, this adjustment permits machines to carry any percentage load at its rated speed, in parallel with other machines.

The operator should make himself familiar with the governors and their associated relay systems for the machines in his charge, as these vary from manufacturer to manufacturer.

Emergency Governors

All steam turbines are equipped with "runaway" or emergency governors which come into operation when for any reason the turbine speed reaches 10 per cent. $\pm$ 1 per cent. of the rated speed. They generally function to shut off the steam supply to the turbine.

Emergency governors are generally of the eccentric-ring or the eccentric-bolt type, and are mounted on the turbine shaft. Minor variations in construction and adjustment occur with the different makes.

STEAM TURBINE MAINTENANCE

The maintenance work on a steam turbine results from general wear and tear in operation and also from mal-operation, and the need for such work shows up from a variety of symptoms.

Checking Performance by Log Sheets

On all machines log sheets as comprehensive as possible should be taken while the machine is in service, and the relevant performance details entered hourly. Then, at monthly intervals, the scope covered by these log sheets should be made more comprehensive and compared with a log sheet taken under similar conditions during the early life of the machine, when it was known that the machine was in perfect operating condition. Deviation from the original log would indicate the need for maintenance.

Yearly Inspection

It is customary, after the first year in service, for a machine to be opened up for inspection, and the condition at this inspection should be a guide as to the frequency of future inspections, bearing in mind such well-known facts that low-pressure blading shows the greatest wear in the first few months in service, after which the wear diminishes, becoming almost static in two or three years. Such bladings are known to be perfectly serviceable after fifteen to twenty years in service, i.e. for the normal life of a steam turbine.

Adjustment of Clearances

At each opening up, a very careful record should be made of the "clearances" throughout the machine, and these should be compared with the original clearance records. The necessary adjustments will become obvious. Increased clearances at packings and glands result in inefficiency and should be re-adjusted.

All drains should be thoroughly examined and cleared if necessary.

Increased bearing clearances and loss of alignment will generally show up as vibration, and should be attended to.

The less frequently the main joints of a machine be disturbed the better, unless it be to rectify a leaking joint.

Blade Washing

Steam turbines on continuous load frequently show, after a period, a falling off of maximum capacity, together with a rise in stage pressures indicating a "silting" up of blading or loss in nozzle area.

This is quite a common experience, and is due to deposits from the steam where considerable feed-water treatment is called for. Generally, the full capacity of the machine is easily restored by the process known as turbine or "blade washing." Most of the deposits are water soluble, and a variety of methods has been devised to introduce the necessary water for washing.

When such a condition arises, it is best to consult the manufacturer as to the method most suitable for the particular machine, as neglect would not only result in loss of capacity but could cause damage through thrust-bearing failures.

This silting-up trouble is not experienced on machines which are frequently stopped and started, nor on machines operated with steam temperatures below 550° F.

Trouble Caused by Blade Shedding

Loss of performance, coupled with increased stage pressures, may also be caused by the loss of some blades at one stage or another, the broken blades passing to succeeding stages, and deforming them, thus restricting the passages to steam flow. Such an occurrence may result in vibration, though blades have been shed without the operators being aware of the fact.

VIBRATION

A well-constructed turbine should run with but little observed vibration, and various experts have from time to time suggested standards as a means of assessing the degree of vibration present on a machine.

For the purpose of comparison with the suggested standards, the measurement of vibration should be in terms of displacement or velocity—preferably velocity, as then consideration is given to the speed of rotation. The following table may be taken as a guide:

CLASSIFICATION OF TURBINE VIBRATION

Classification of Vibration	*In Terms of*	
	Displacement (in.) Amplitude at 3,000 r.p.m.	*Velocity. R.M.S. in./secs.*
Excellent	< 0·0005	< 0·05
Very good	0·0006–0·0012	0·1
Normal	0·0013–0·002	0·175
Slightly rough	0·002 0·003	0·225
Excessive	0 003 and upwards	0·25 and upwards

A good turbine operator will become familiar with the degree of vibration on his charges, and will be sensitive to deterioration in smooth running. Changes for the worst in running indicate the need of maintenance, and in case of doubt the manufacturer should be consulted, for the possible sources causing vibration are so varied that to diagnose them correctly would need the services of an expert, with possibly a battery of instruments to assist in the diagnosis.

The figures referred to in the preceding table should apply to the vibration as measured at bearing pedestals and housings, and not to odd parts of the machine.

E. D. E.

STEAM TURBINE MAINTENANCE

The maintenance work on a steam turbine results from general wear and tear in operation and also from mal-operation, and the need for such work shows up from a variety of symptoms.

Checking Performance by Log Sheets

On all machines log sheets as comprehensive as possible should be taken while the machine is in service, and the relevant performance details entered hourly. Then, at monthly intervals, the scope covered by these log sheets should be made more comprehensive and compared with a log sheet taken under similar conditions during the early life of the machine, when it was known that the machine was in perfect operating condition. Deviation from the original log would indicate the need for maintenance.

Yearly Inspection

It is customary, after the first year in service, for a machine to be opened up for inspection, and the condition at this inspection should be a guide as to the frequency of future inspections, bearing in mind such well-known facts that low-pressure blading shows the greatest wear in the first few months in service, after which the wear diminishes, becoming almost static in two or three years. Such bladings are known to be perfectly serviceable after fifteen to twenty years in service, i.e. for the normal life of a steam turbine.

Adjustment of Clearances

At each opening up, a very careful record should be made of the "clearances" throughout the machine, and these should be compared with the original clearance records. The necessary adjustments will become obvious. Increased clearances at packings and glands result in inefficiency and should be re-adjusted.

All drains should be thoroughly examined and cleared if necessary.

Increased bearing clearances and loss of alignment will generally show up as vibration, and should be attended to.

The less frequently the main joints of a machine be disturbed the better, unless it be to rectify a leaking joint.

Blade Washing

Steam turbines on continuous load frequently show, after a period, a falling off of maximum capacity, together with a rise in stage pressures indicating a "silting" up of blading or loss in nozzle area.

This is quite a common experience, and is due to deposits from the steam where considerable feed-water treatment is called for. Generally, the full capacity of the machine is easily restored by the process known as turbine or "blade washing." Most of the deposits are water soluble, and a variety of methods has been devised to introduce the necessary water for washing.

Levelling and Fixing on Foundation

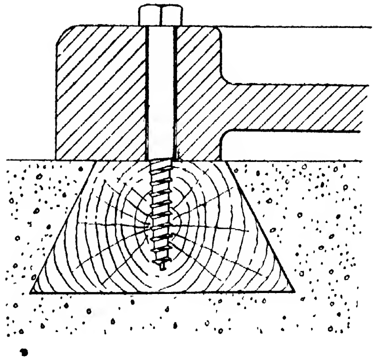


FIG. 1.—METHOD OF PROVIDING HOLD FOR COACH-SCREW IN CONCRETE FOUNDATION

Levelling is effected by means of wedges and fixing by coach-screws or bolts. When concrete forms the support, screws can be run into wood blocks embedded in the concrete, as seen in Fig. 1, but many of the heavier machines are never bolted down, the only thing done being to place stop-pins at suitable locations just to prevent the possibility of shifting. Simple wood or steel wedges are utilised to adjust beds and frames on their concrete foundations, but the tendency is towards providing more precise devices with the machines, either lugs with vertical screws or screw-operated wedges.

Levelling Screws

When screws are fitted they press on steel plates sunk a slight distance, and after correct level has been effected, the underside of the bed is run up with grouting. Fig. 2 explains the arrangement along the foot of a bed for a Pearn-Richards universal boring, facing, and milling machine. Large T-slotted plates, as employed on travelling-column horizontal-spindle boring machines, have the levelling screws sunk below the top surface of the plate, with squared end for manipulation by box spanner.

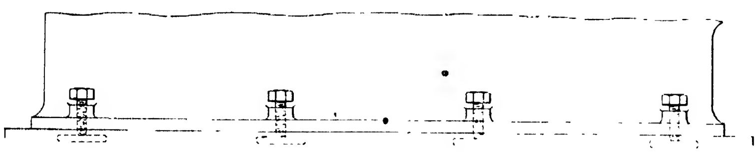


FIG. 2.—SYSTEM OF LEVELLING BORING-MACHINE END
Using screws, grouting being afterwards run under the base.

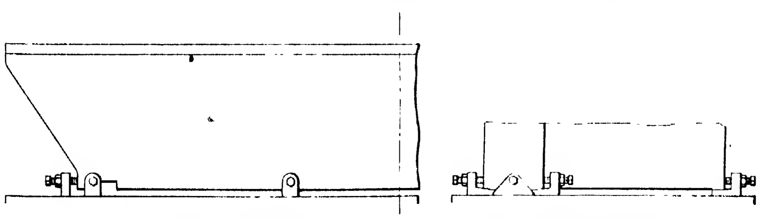


FIG. 3.—HALF-ELEVATION AND END VIEW OF GRINDING-MACHINE BASE
Showing levelling wedges and end stops.

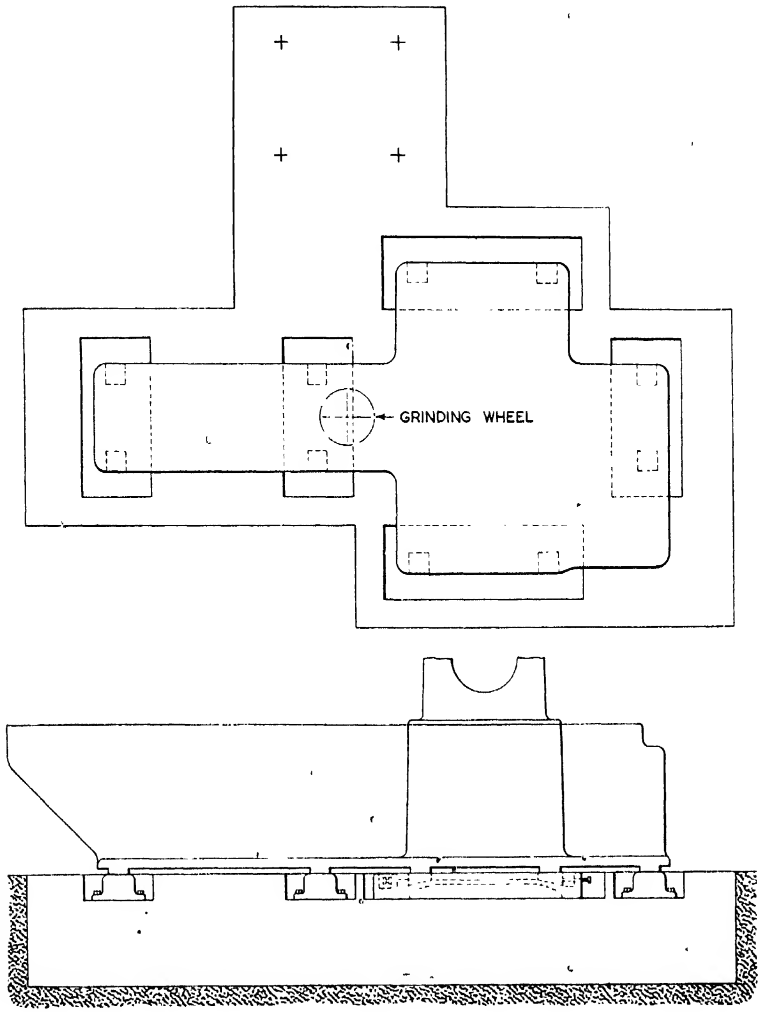


FIG. 4.—LAYOUT OF SET OF WEDGE BLOCKS TO LEVEL CHURCHILL PLANO-GRINDER
When adjustment of wedge blocks in the pits is completed, they are grouted in position.

Wedge Adjustments

Wedges include some connected to the base, others independent. The former (Fig. 3) feed forward or backward by operation of the screws passing into the

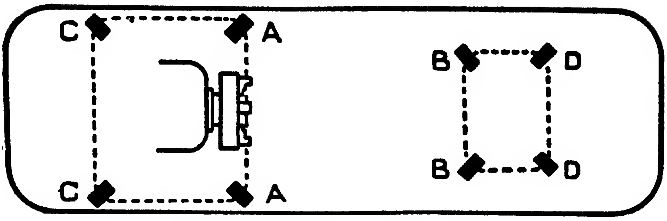


FIG. 5.—LEVELLING A TURRET LATHE

The spirit level is first placed across the bed of the machine near the headstock. Iron wedges *A* are then driven under the inner corners of the feet at the headstock end of the lathe. Adjust the wedges until a correct reading is given on the spirit level. Place the spirit level across the opposite end and level up by means of wedges *B*. Take a further reading at the headstock end to see that the alignment has not been disturbed. When the bed is perfectly level, wedges *C* and *D* should be lightly tapped underneath to give support. The spaces between feet and foundation should then be grouted up.

base. Independent wedge outfits, necessary for planing, milling, grinding, and other machine beds, consist of units each comprising a substantial sole-plate, and a wedge sliding thereon, by one screw, or a couple, fore and aft. Graphite between the meeting faces will ensure a smooth steady motion, permitting adjustment to be made without jerkiness. The layout of a set of levelling blocks for a large plano-grinder by the Churchill Machine Tool Co., Ltd., appears in Fig. 4. The three sets under the bed each have two wedges like those under the standards. The screw heads are protected within pockets, and locknuts prevent loosening.

Pits are cast in the concrete setting, as is evident from the plan view, to receive the wedge blocks, and after levelling, all the blocks may be grouted in position. The driving motor stands at the left, the bolt centres being indicated.

Tests for Accuracy of Setting

Tests for truth of levelling vary according to the kind of datum surfaces which are present. A simple bed can receive the spirit level direct, or on a straightedge if gaps must be spanned. When there are Vs, as in a planing or grinding machine, the top edges adjacent thereto may have been planed at the same setting, and a straightedge can be laid over these edges upon which to rest the level. Generally, however, two ground bars of uniform diameter are laid in the Vs. The straightedge can then be placed on the bars.

Longitudinal condition is tested with a straightedge laid along the Vs, and level on top of it; or a wire is strained between supports attached to the ends of the bed, midway between the Vs, and a special gauging indicator fixed on the straightedge, which rests upon the ground bars, tried along the wire successively the whole length.

Testing for Settlement after Installation

These tests must be made occasionally, after installation, as the shocks and severe duty of a planing machine are liable to induce settlements. It should be

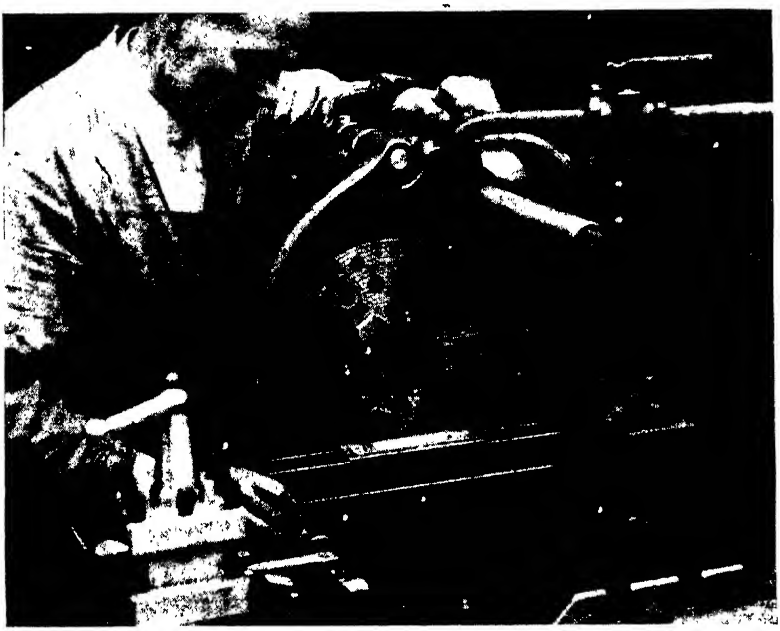


FIG. 6.—TESTING MACHINE BED FOR ALIGNMENT

mentioned that if the machine has been set up properly, the table will obviously be true, as it was planed in the maker's shop; it is a mistaken policy to plane it over after erection, because if such a practice is found necessary, the trouble lies in the foundation.

Installation of Motors

A great deal of detail formerly associated with belt drives has been eliminated, especially in respect of large machines, the driving being done by one or more motors. The installation of these often needs no attention, as the machine base or frame carries them; if mounted a little distance away on the floor, there is not much difficulty. Transmission may be by belt, the tension of which may be adjusted by sliding the motor along its rails, or a jockey pulley may be arranged on the outside of the belt to give tension, this applying particularly to short drives. Direct connection by shaft, preferably with a flexible coupling to reduce shocks and compensate for slight faults in alignment, has largely superseded belting, while spur or helical pinion or worm gears are also much used.

Countershafts

Countershaft arrangements involve more care in setting up, to ensure parallelism so that belts will track properly.

This kind of location is a matter of direct measurement from the countershaft to the machine bed; or strings can be suspended from near the ends of the shaft, and measurement taken along horizontally to the machine. Levelling of a countershaft may be very nearly correct if beams or stringers are fairly true, only slight adjustment by shims under hangers being necessary.

It is best to deaden sound by interposing some kind of insulator between the feet and the girder, such as wood, leather, canvas belting, or rubber. Convenience of mounting may be achieved by employing clips which can be slid along to any spot and facilitate alterations. Cross girders for supporting hangers are likewise attached to main beams in a similar manner. Special care should usually be observed to use the specified sizes of driving pulleys from the lineshaft. If pulleys of an incorrect size are used, not only will the speed of the grinding wheels be affected, but the speed and feed tables cast on or fastened to the machine will be rendered false and misleading.

Selection of Belts

Proper selection of belts is also very important, both as to straightness, flexibility, and good jointing. Some types of machines are much affected by bad belts failing to deliver good results, this being most noticeable on grinders, where vibration causes chatter marks. Readers may like also to refer to the article on the "Installation of Electric Motors and Auxiliary Equipment" (page 104), and to the article on "Power Transmission by Belting" (page 458) in this volume.

MAINTENANCE AFTER INSTALLATION

Bearing in mind that a machine tool has to reproduce accurate cylindrical, plane, and varied surfaces, the maintenance is much more than mere attention to lubrication, cleanliness, and avoidance of bruising or fracture.

Changes due to Settlement and Wear

Changes, some rather subtle, develop, and the maker has to design as well as possible to minimise undesirable results therefrom. The best materials, rigid construction, and excellence of fitting have the effect of delaying trouble. In the first place a strong base or bed or framing is imperative for the following reason. Any settlement in the foundation, or weight of work, jigs, fixtures, etc., immediately causes distortion in a comparatively weak structure. This not only affects adversely the accuracy of machining, but soon depreciates the principal elements by making them wear irregularly. Spindles tend to rub more at some sections of the bearings, and knees, tables, or slides bear hard across corners, or at certain spots along ways.

Wearing Surfaces

Wearing surfaces are now also arranged in better fashion, with more extensive contact, and good protection against the ingress of cuttings, dust, grit, and foreign matter generally.

The familiar narrow guide likewise helps in delaying alignment errors, as it gives accurate control, near to the site of the chief action of the machine.

Beds and Slides

Extra hard or chilled beds and slides form a valuable kind of wear-resisting scheme; some firms adopt hardened and ground-steel bars or "wearing" plates, attached to bed tops and slides, if the duty happens to be very severe.

With regard to lathe beds, the practice of having these hard preserves alignment, since the saddle, being of softer metal, cannot readily wear the bed more at one place than another—near the head, for instance.

Rectification of Errors in Vital Parts

Several methods are pursued to rectify or compensate for errors when they do appear. The more drastic treatment of replaning beds and other items, and reboring shaft and spindle bearings, is not resorted to until affairs become very bad, or a machine has to be reconditioned for sale.

Scraping

Skilful scraping will do a lot (besides correcting worn or scored bearings), such as tilting a headstock to bring the spindle into perfect alignment, or throwing over a slide for parallelism, or tipping a standard either way.

Height may also be varied by scraping, to bring any member down slightly to suit an altered condition.

This kind of location is a matter of direct measurement from the countershaft to the machine bed; or strings can be suspended from near the ends of the shaft, and measurement taken along horizontally to the machine. Levelling of a countershaft may be very nearly correct if beams or stringers are fairly true, only slight adjustment by shims under hangers being necessary.

It is best to deaden sound by interposing some kind of insulator between the feet and the girder, such as wood, leather, canvas belting, or rubber. Convenience of mounting may be achieved by employing clips which can be slid along to any spot and facilitate alterations. Cross girders for supporting hangers are likewise attached to main beams in a similar manner. Special care should usually be observed to use the specified sizes of driving pulleys from the lineshaft. If pulleys of an incorrect size are used, not only will the speed of the grinding wheels be affected, but the speed and feed tables cast on or fastened to the machine will be rendered false and misleading.

Selection of Belts

Proper selection of belts is also very important, both as to straightness, flexibility, and good jointing. Some types of machines are much affected by bad belts failing to deliver good results, this being most noticeable on grinders, where vibration causes chatter marks. Readers may like also to refer to the article on the "Installation of Electric Motors and Auxiliary Equipment" (page 104), and to the article on "Power Transmission by Belting" (page 458) in this volume.

MAINTENANCE AFTER INSTALLATION

Bearing in mind that a machine tool has to reproduce accurate cylindrical, plane, and varied surfaces, the maintenance is much more than mere attention to lubrication, cleanliness, and avoidance of bruising or fracture.

Changes due to Settlement and Wear

Changes, some rather subtle, develop, and the maker has to design as well as possible to minimise undesirable results therefrom. The best materials, rigid construction, and excellence of fitting have the effect of delaying trouble. In the first place a strong base or bed or framing is imperative for the following reason. Any settlement in the foundation, or weight of work, jigs, fixtures, etc., immediately causes distortion in a comparatively weak structure. This not only affects adversely the accuracy of machining, but soon depreciates the principal elements by making them wear irregularly. Spindles tend to rub more at some sections of the bearings, and knees, tables, or slides bear hard across corners, or at certain spots along ways.

Wearing Surfaces

Wearing surfaces are now also arranged in better fashion, with more extensive contact, and good protection against the ingress of cuttings, dust, grit, and foreign matter generally.

are available special pads of cork or other material which are placed under and round the normal concrete foundation, or this may be arranged to rest on, or be suspended on, a series of springs. In both cases excellent results have been achieved.

Most hammers are secured by foundation bolts, which should not be permanently fastened into the concrete but arranged in timber boxes, with special plates at the bottom, so that in the event of breakage the bolt may be withdrawn and replaced.

Forging hammers should have the anvil block set up above the correct working position to allow for sinking in the first few months of working. A safety mark is normally provided on the hammer to indicate when the lowest safe working position has been reached, either as a result of sinking of the block, or by reason of wear of the pallets.

Trouble with an otherwise good foundation is sometimes caused by scale from the forgings getting down and working under the block. With drop hammers this is usually prevented by filling in all round the block with a weak mixture of concrete (which can be broken up if necessary), or with earth. In the case of forging hammers, the trouble can sometimes be prevented by arranging a sheet-steel cowl round the block above the opening in the baseplate. Alternatively, gib-headed wooden packing pieces may be inserted between the baseplate and anvil block.

Forging Hammers

PALLETS.—It is very important that the pallet faces should be kept true, as considerable strain is otherwise thrown on to the piston rod. If the faces become uneven they should be trued up by machining or grinding. Small pallets made in 0·6 per cent. carbon steel usually have the faces hardened, and if they have to be softened this can be done by heating to about 860° C. and allowing to cool gradually. To reharden, heat to 860° C. and quench the face in cold running water to a depth of about half an inch, leaving the pallet in position until cold. No tempering is required. There is, of course, always some risk in this softening and rehardening both of cracks developing and of the dovetail being warped. The latter point should be watched and the dovetail cleaned up if necessary. It is essential that the dovetail should be true and the key a good fit along its whole length. When inserting the key put oil or grease on it to facilitate removal later. Badly fitting keys are often a source of anvil-block breakages.

ANVIL BLOCK.—If the dovetail of a block should be broken off, it is usually possible to effect an excellent repair by cutting down the block to a considerable extent and fitting a separate steel top. This should fit over a dovetail projection machined on the block, and be secured by a key, i.e. the recess should be in the steel top, not in the cast-iron anvil block.

PISTON RODS.—Piston-rod breakages will occur, but they can be reduced by attention to the pallets, as mentioned above, avoiding excessive hammering on the pallet edges, keeping the slides properly adjusted, and by care when

starting up. Heavy hammering should be done as near the centre of the pallets as possible, and unnecessary hammering on thin, cold material should be avoided. If the nature of the work inevitably throws extra side thrust on to the tup, as, for instance, when drawing down taper packings (or slips), lengthened slides are a great advantage. When starting up, particularly in cold weather, it is a wise precaution to lay a piece of hot metal on the top of the tup alongside the rod before work begins.

SLIDES.—Where hammers are provided with slides, these are nearly always adjustable, and should be kept adjusted so as to give the tup the maximum support. When the tup is pushed hard up against both surfaces of one slide, there should be a maximum of 0·018 in.– 0·028 in. clearance, according to the size of hammer, between each of the other slide faces. This is specially important in the lower part of the stroke, and it is an advantage to have a slightly larger clearance in the upper portion. The slides should be kept well greased, and, if desired, grease-gun nipples may be fitted to the tup delivering grease between the tup and slides.

PISTON RINGS.—One of the most frequent causes of hammers working feebly or irregularly is lack of attention to piston rings. Worn rings allow leakage, and a broken ring may do considerable damage to the cylinder. Rings should therefore be examined regularly and replaced if necessary. This can be done by removing the cylinder cover and lifting the piston head above the top of the cylinder; and it is usually not necessary to remove the rod from the tup. The rings should be pegged to prevent rotation. In pneumatic hammers the pump rings are often of cast iron, and additional care is therefore demanded in fitting. To remove the pump piston, the connecting-rod big-end bearing must be taken down. When replacing the pump piston, the trunk rings can be held in position by thick grease or by a piece of string which is pushed up as the trunk enters.

Removing Tup from Rod

In most hammers with slides the piston rod is tapered at the lower end to fit a conical hole bored in the tup. Sometimes a copper liner is inserted between rod and tup. If it is necessary to remove the tup from the rod, this should be done before dismantling the hammer, as follows: drive out the tup pin, remove the pallet, and insert in the hole in the tup under the rod a steel punch of such length as to keep the pallet when replaced about ¼ in. clear from the face of the tup. Key up lightly, and strike one heavy dead blow, which is usually sufficient. If difficulty is experienced in loosening the tup, heat it back and front and proceed as above, before the heat has time to penetrate to the rod. Great care must be exercised in raising the rod, when separated, to prevent it striking the cover.

Some makers use a ball-and-socket arrangement for securing the tup of large hammers, the ball end of the rod being held between steel cup pieces, above and below which are alternate packings of hardwood and steel. The whole is secured in the tup by cotters or by a dovetailed cap. The cotters should

be driven well home and secured by a steel pin or ring at the small end. The cap is secured by a steel key or keys, but requires forcing down on to the packings before the key can be inserted. Screws are usually provided for this purpose.

Steam and Air Hammers

Any steam hammer may be driven by compressed air, if desired; but as the working temperature is much less with air, the clearances can be reduced. As a rough guide it may be taken that the piston clearance in an air hammer should be about 0·0005 in. per inch of diameter, and for steam hammers 0·001 in. per inch of diameter.

STEAM HAMMER JOINTS.—These should be watched, as a leaky joint will result in feeble working. Steam leaks are visible, but air leaks are not. To repair these, a special jointing material is used. Where hammers are fitted with a separate valve chest a hole is usually drilled into the cavity between the top and bottom ports, so that should the joint between the cylinder and valve chest break, steam will escape and show up the fault. The jointing material must extend, not only round the flange, but also across the face so as to separate the two ports from the cavity.

ARRANGEMENT OF STEAM PIPES.—In fixing steam piping, care should be taken that it falls in the direction in which the steam travels. A steam trap should be arranged at the end of the steam main. All branch pipes should be taken from the top of the main and the last branch should be 3 ft. or more from the end of it. It is convenient to have flanged joints near to the hammer, and pipes there should not be too rigidly fixed or should be flexible.

DRAIN PIPES.—Arrange a valve on each of the pipes from the stop valve, control valve, and cylinder before they are joined together, or make very sure that live steam cannot escape to exhaust. Always open up all drain pipes for a few minutes when starting up.

STUFFING BOXES.—Makers usually recommend a fairly soft packing, but some of the proprietary brands can be used with success. In repacking take care to tighten up the gland evenly all round, and use only sufficient pressure to prevent leakage.

LUBRICATION.—A lubricator, either hand operated or automatic, is normally supplied by the maker. A good cylinder oil should be used, of a type that will emulsify. For air-driven hammers a machine oil is recommended. There should be a back-pressure valve between the cylinder and lubricator.

VALVE GEAR.—Wear inevitably takes place in the quickly moving parts of the valve motion, and in time the accumulated lost motion becomes excessive and the valve setting is affected. It then becomes necessary to fit new pins in all joints.

Pneumatic Power Hammers

Much of the information given above applies equally to these electrically driven hammers. Special points are dealt with below.

STUFFING BOXES.—Hammers with normal piston rods and separate tups

ordinarily have a stamped-steel stuffing box containing a strong spring and separate neck-ring and collar bush of phosphor-bronze. The packing, which should be of the soft variety, is automatically tightened by the spring. When necessary, an additional ring of packing can be inserted to make up for wear. Hammers of the "clear space" type, with a large-diameter ram, have separate steel packing pieces which bear against the flats on the ram. As wear takes place these must be packed out with shims. Asbestos-tape packing strips are arranged in the stuffing box, but the principal function of these is to prevent dirt being drawn up into the box and to ensure that the ram is properly lubricated. Reliance is placed on the fit of the stuffing box to prevent leakage.

LUBRICATION.—Some hammers are provided with hand-pump lubricators as mentioned above, but the more modern designs have automatic lubrication from a sump arranged in the baseplate. In such cases a sight-feed is provided on the pipe to the cylinders, and this should be filled with water, or preferably glycerine. The screw adjustment for flow should allow the passage of just sufficient oil to keep the ram slightly oily. The oil filter, if provided, should be used at least once a day. In water-cooled hammers there is a tendency for water to collect at the bottom of the sump, and this should be drained away regularly.

VALVES.—These do not normally move with each blow of the hammer like a steam-hammer valve, but are provided with two or more one-way valves of the disc type. If a hammer begins to work erratically, a sticking or broken disc should be suspected.

DRIVES.—It is important that the voltage shown on the motor is actually available at the terminals, as otherwise the motor may slow up or overheat. Fuses and overload releases should be set to carry about 1½ times the full-load current, and if the supply is A.C., time lags should be provided.

Friction Drop Stamps

The mechanism of these machines is simple, the parts are easily seen, and there should be little to trouble the millwright. It is of primary importance that adjustments when necessary should be made promptly. The shaft bearings are usually ring-lubricated, but lifter-arms require the use of the grease gun, maybe twice a day. The friction band should be free on the drum when at rest, and the length of the lifting belt should be such that with the dies just touching the arm may be lifted about 2 in. from the buffer. The lining of the friction band can be renewed when worn, and it is wise to have a spare band which can be put into use while this relining is done. The surfaces of the tup, dieholder, and anvil block must be remachined if they become badly indented by use. When adjusting the tup guides, allowance must be made for the expansion of the tup as it heats up. If automatic gear is provided, the adjustments of the pulling cord and lifting belt must be watched with extra care, to avoid snatching. If work has to be done on the dies, the tup should be propped up on a baulk of timber—not on the steel hold-up prop.

D. L. P.

POWER TRANSMISSION BY BELTING

ONE method of transmitting power to a machine by mechanical means is to use a belt drive. For this purpose either flat belt or V-belts may be used. The latter is really a modification of the flat belt, and was originally developed to meet the need for large power drives then unobtainable with flat belts. However, modern treatments of leather have produced flat belts suitable for these drives. Both types may be made from either leather, canvas, or rubber, depending on the form of drive required.

FLAT BELTS

In addition to the ordinary type of leather belting, a belt made from links of leather has been developed. The links are fastened together in pairs by steel pins, the belt running on the edge of the link, and the grain of the leather lying at right angles to the face of the belt. This double link construction increases the strength of the belt. When the pulleys to be used are cambered, the belt is made with a flexible centre, which allows the full width of the belt to contact the pulleys.

Balata belts are usually canvas or cotton belts impregnated with balata, which renders them specially suitable for heavy duties.

To join or make endless a flat belt, one of the following methods may be used, namely, lacings (principally used on leather belts), metal fasteners of various kinds, and cemented or solutioned joints. Excellent results can be obtained from both metal-fastened and laced joints, but the cemented joint, when properly made, constitutes what is really an endless belt, and eliminates the disadvantages to which a badly made metal or laced joint can be subject.

Cemented or Endless Joint (Balata Belts)

To make an endless joint, the first thing is to decide on the length of lap required. The following data gives the sizes for balata belting:

From 1 in. to 2 in. wide up to 10 ft., 6-in. lap.

Over 10-ft. lengths up to 12 in. wide, 12-in. lap.

Over 12 in. wide, length for breadth, i.e. 14 in. wide, 14-in. lap, etc.

To splice a 5-ply balata belt with a 12-in. lap, first pare round edge back to 12 in. The covering matter is then scraped from the belt ends back to the mark or butt of splice. The belt end is then heated gradually at a stove or by means of

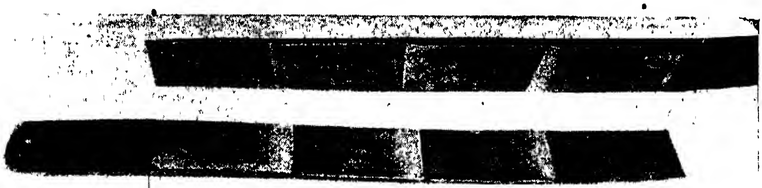


FIG. 1.—SPLICE FOR 5-PLY BALATA BELT

With a lap of 12 in. Showing how the two ends are stepped for cementing together to make an endless belt. Note how the second top ply is cut back to within ½ in. of the butt, the top layer being cut to within 3 in. of the butt.

heated irons to soften it for opening up. The top ply is removed to within 3 in. of the butt, afterwards the second ply is removed to within ½ in. of the butt (in every case of multiple plies the second ply is removed in this manner). The third ply is removed to within 6 in. of the butt, and the fourth ply to within 9 in. The edges of the steps should be pared down to avoid any abruptness in the finished joint, and the surface of the steps roughened to allow solution to penetrate the fabric.

When both ends of the belt have been prepared as above, they will be ready for the solution, which can be obtained from belt manufacturers. The solution must be brushed well into the fibre with a stiff short-haired brush once lightly and once heavy, thus giving two coats. The belt should now be allowed to stand for about thirty minutes to enable the solvent to evaporate. The ends should then be heated for a second time, and the ply marked 4 is stuck down over the ½-in. piece where the ply was removed. The joint is now ready for final heating, which must be gradual and not above 150° F. in case the solution catches fire. The ends are now pressed down carefully and squarely and placed in a press, which must be first wetted to prevent the belt from sticking. The splice should remain in the press for about ten minutes, after which it may be removed and allowed to cool. The edges can then be carefully pared and rounded over with a warm knife. No ragged edges must remain, and the covering should be dressed over the raw edges.

Cemented or Endless Joints (Leather Belts)

In order to splice a leather belt, the use of a spokeshave is necessary. With this the leather is tapered away from the butt or splice mark to the end of the belt. It is better to do the rough work with the spokeshave and then trim with a scraper or very sharp knife. The taper should be true and gradual, and there must be no lumps behind the butt.

The glue used must be hot, and should be worked well into the belt. The two ends are then brought together, rubbed with a slight pressure, and then placed in a clamp or press, where it should remain until the glue is thoroughly set.

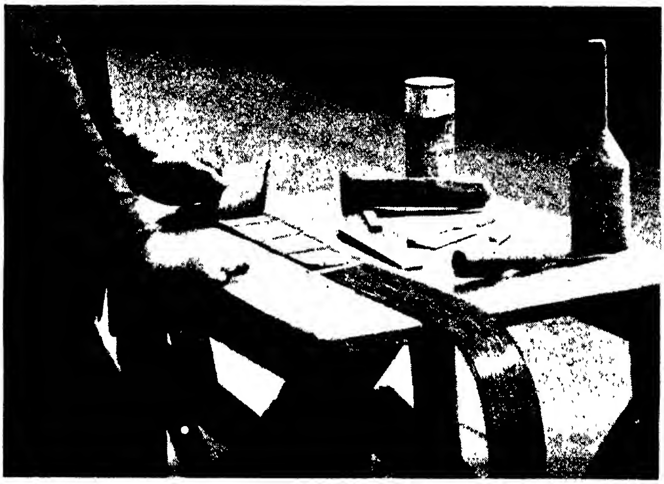


FIG. 2.—SPLICING BALATA BELT

Showing how second top ply is cut back to ½ in. of the butt. The worker is holding the longer top ply with his left hand.

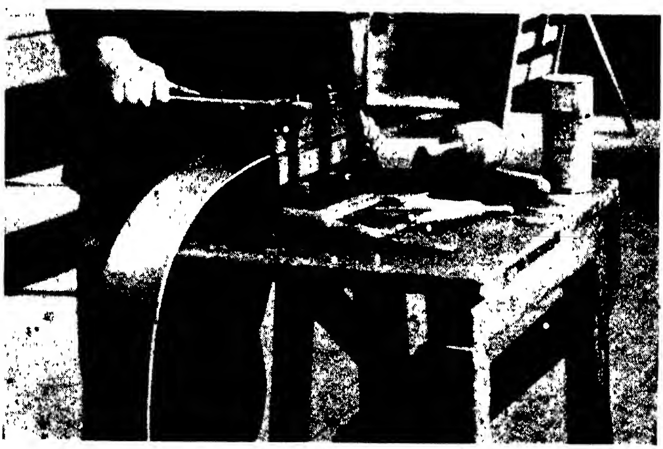


FIG. 3.—SPLICING BALATA BELT

Pressing the ends in press block after solutioning and heating. Block must first be wetted to prevent belt from sticking to it. Allow belt to remain in press for ten minutes.

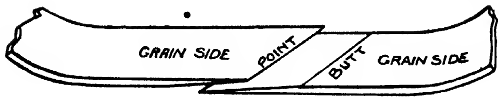


FIG. 4.—ENDLESS LAP FOR SINGLE LEATHER BELTS

Belt is tapered at each end. The ends are then joined with glue or cement.

Two coats of glue should be given. Alternatively, the ends may be joined, using waterproof cement, which may be applied cold.

The following table gives the minimum amount of lap required, although alterations may have to be made to suit particular service conditions:

Width of Single Belt in in.	*Length of Lap in in.*
1–4½	4½
5	5
6–8	6
9	6½
10–14	7
15–24	8

All double belts should have a 6-in. lap at least.

Metal Fasteners

There are many kinds of metal fasteners, and there are types for every kind of service. If a good type of fastener is selected, the belt joint should not give any trouble.

When cutting a belt for a metal fastener or for a laced butt joint, a metal square should be used. This is to ensure that the cut is at right angles to the

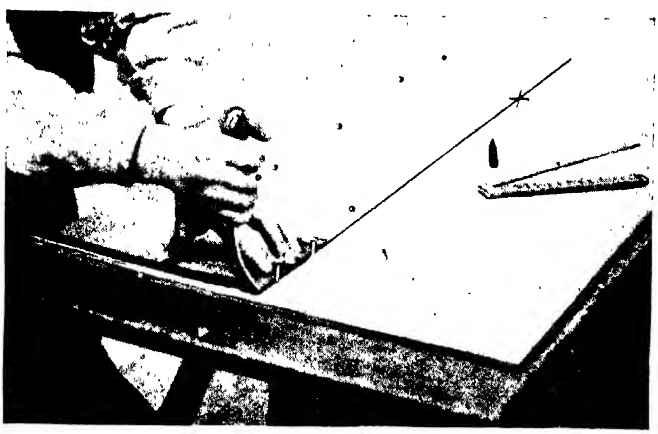


FIG. 5.—METHOD OF CUTTING BROAD BELTS

Using a short metal square. Note small nails on centre line to steady the square.

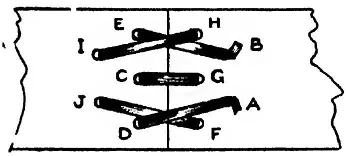


FIG. 6.—BELT LACING

centre line of the belt. If a square is not employed, one or both ends are liable to be cut unevenly, and these conditions cause the belt to slip from side to side of the pulley and impose irregular strains which will shorten the life of the belt. The knife blade should be made wet occasionally to make it cut easily.

Sometimes it is difficult to cut a wide belt square on account of slight variations in the width of the belt, and because the sides may not be parallel. If this is so, the following method should be satisfactory.

Find the centre of the belt near the point to be cut, then at a point 2 ft. or 3 ft. back from this dimension locate the centre again and between these two points draw a clean sharp line. Next hold the leg of the square against the centre line just made, and trim off one half of the end of the belt by cutting along the other leg. By laying the square on the other side of the centre line the other half can be cut. The square should be firmly held in position while this is being done, and if two small nails are driven in the belt on the centre line, they will help to keep the square from slipping.

Lacing a Belt

In punching a belt for lacing, it is desirable to use an oval punch, the longer diameter of the punch being parallel with the belt, so as to cut off as little leather as possible. Two rows of holes should be made at each end of the belt. The punched hole should be smaller than the lace width, so that the lace can bend and fill the hole. Do not punch a hole larger than is absolutely necessary to draw in the lace. Begin to lace in the centre of the belt, and care should be taken to keep the ends exactly in a line and to lace both sides with equal tightness. The lacing should not be crossed on the side of the belt that runs next to the pulley.

The size of laces for various widths of belting should be as under:

Width of Belt in in.	*Width of Lace in in.*
24 and over	$\frac{1}{2}$
6–24	$\frac{3}{8}$
2–4	$\frac{5}{16}$
2 and less	$\frac{1}{4}$

Pulleys

Pulleys are made in various designs and of different materials, according to the speed at which they must run and the power to be transmitted.

In some instances, generally owing to the speed of the driven machine, it is necessary to have comparatively large pulleys, although only a small power is to be transmitted, and for such cases wood or pressed-steel pulleys are useful, as they are light and cheap.

For medium and heavy drives cast-iron pulleys are standard practice, ordinary designs are generally suitable for rim speeds up to 4,000 ft. per minute. For higher speeds (up to 5,500 ft. per minute) it is usual to make the pulley so that it can be accurately balanced.

FIG. 7.—PUNCHING LEATHER BELTS FOR LEATHER LACING

Oval punch holes parallel with belt.

When several pulleys are placed in a shaft, additional load is put on the bearing, and this is an important point in favour of wrought-iron pulleys, which are comparatively light and strong. The usual design of wrought-iron pulley has a cast-iron hub with a wrought-iron rim; pulleys up to 10 in. wide have one row of arms, wider pulleys may have two or three rows of arms, according to duty. For speeds over 4,000 ft. per minute and for arduous duties it is usual to adopt a form of treble-riveted construction, which gives more stiffness to the rim. Small high-speed motor pulleys are generally made of cast iron and of plate centre construction.

Pulleys should be wider than the belt and the rims slightly convexed; a pulley 12 in. wide should be increased in diameter at the centre not more than $\frac{1}{16}$ in. Fast and loose pulleys should have slightly crowned faces, but the pulleys from which they receive their drive should be flat.

Flat-belt Drives

Through the courtesy of Messrs. John Tullis, Ltd., we are able to reproduce, on page 466, formulæ which may be used for calculating the length of belt required for various types of flat drive and also the data required for calculating pulley sizes and horse-power transmitted by flat belts.

The individual short-centre drive is one of the most popular drives used to-day. In nearly all cases of short-centre driving there is usually a high ratio between the two pulleys, and the larger the ratio the heavier the belt which can be employed. It has been found that the belts used on these drives transmit the power more uniformly and efficiently and are also less costly to install and to maintain. Losses due to the decreased arc of contact on the small pulley can be largely compensated for by increasing the thickness of the belt, and fully covered by increasing the width. The thickness of the belts used vary from 6 mm. to 12 mm. according to the diameters of the pulleys and the horse-power

required to be transmitted. The uniform tension at which flat belts operate gives an advantage over multiple drives, which are exceedingly sensitive to tensions and seldom work at the same tension owing to uneven stretch.

Another type of transmission system is the modern group drive. With this arrangement it is possible to drive from two to ten machines from one motor instead of individual drives for each. The size of a motor for an individual drive is determined by the peak or starting load. In every case a motor must be large enough to maintain the speeds required by production at maximum load without overload. In modern group driving, owing to the fact that peak or starting loads rarely overlap, the motor carries only the average load. Maintenance and installation costs of an individual drive far exceed those of the group drive.

Care and Maintenance of Belts

A reliable belt dressing, used sparingly but frequently, will keep the belt in good condition. The use of preparations which leave lumps on the belt or pulley is to be deplored. Care should always be taken to prevent oil getting on the belt, as this will cause loss of power by slip and is particularly harmful to balata or cotton belts, tending to separate the plies. An application of ground chalk will absorb the oil on a leather belt and make it workable for a time.

In order to get the best use from a belt drive correct length and arrangement are essential. In the case of a balata belt the canvas side is the driving face, whether the drive is crossed or open. The correct driving side for leather is usually the grain side, but adequate belt dressing will enable the flesh side to transmit power equally as well. In the case of a good leather belt the grain side should transmit nearly twice the power conveyed by the flesh side. The flesh side, which has the greatest tensile strength, will stand the stretching strain necessary in the outside bend around the pulleys. The flesh side being the stronger, must be protected from wear. Rubber belts, especially wide ones, should never be run on to pulleys. For endless belts, the best method is to take out both shafts, place the belt in position on the pulleys, and replace the shafts in their bearings. The maximum speed at which a rubber belt should be operated is 5,000 ft. per minute, but greater life is obtained when the speed is kept under 4,000 ft. per minute.

If a belt is overloaded, the use of a rider belt running loose on top of the main belt will increase the power by about 75 per cent. Rider belts should, however, not be run on small-diameter pulleys. Crossed belts should not be used for short-centre drives.

FAULTS IN FLAT-BELT DRIVES AND THEIR CORRECTION

Belt Running Badly

The alignment of pulleys and/or shaft may be faulty.

One or both of the pulleys may be badly aligned.

If steam is coming from the engine irregularly, a wave motion will be imparted to the slack side of the belt.

A heated bearing—the result of failure to lubricate—will have this effect.

Slipping of Belt

Slip is evidenced by the pulley face becoming polished. In the case of a balata belt, a few drops of castor oil applied to the face of one of the pulleys or the driving side of the belt will probably cure this. It is better to use one of the recognised dressings on the market for leather belting.

If the belt is still found to be slipping, it should be shortened.

Slip may also be due to the belt being overloaded. This may be checked by reference to the manufacturer's horse-power tables, making due allowance for intermittent loads, arc of contact, and working conditions.

The belt may be too heavy for the small pulley.

The pulley centre may be too short, belt contact therefore being lost.

If a longer drive cannot be arranged, a jockey pulley or gravity idler may be installed with advantage.

The belts may be too narrow. A broad, thin belt is always preferable to a narrow, thick belt.

Belts coming off Pulley

It may be running at an excessively high speed.

The alignment of the pulleys may be at fault.

Joint may not have been cut square.

Belt may be overloaded.

Incorrect tension may be the cause.

Fraying at Belt Edge

Flanged pulleys may be the cause; these serve no useful purpose and may be discarded.

If the shifter is being forced over instead of eased over, it will wear the edges and buckle the belt. Roller forks should be used where possible.

If metal fasteners are used to join a belt running on a fork drive, they will strike and barb the forks if they are too close to the edge of the belt. The forks in turn will destroy the edge of the belt.

Loss of Power

A belt put on the pulleys too tight causes unnecessary strain on the bearings, and also on the belt. It will develop less power than if put on at the proper tension. Generally speaking, the more slack in a belt up to slipping point, the more power will be transmitted. This does not apply in all cases, however, as certain drives require a tight belt.

BELTING FORMULÆ FOR FLAT BELTS

The following formulæ and application are supplementary to the engineering service given by manufacturers on specific drive problems:

The symbols used in the formulæ are:

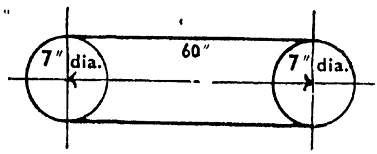
D = diameter of large pulley.
d = diameter of small pulley.
C = distance of shaft centres.
W = width of belt.
R.P.M. = revolutions per minute.
S = speed of belt in feet per minute.
C.E.T. = corrected effective tension.

To find butt length of belt required:

(1) When pulleys are approximately the same diameter:

Add together A and B.

$$A = \frac{D+d}{2} \times 3{\cdot}14\text{; e.g.}$$



$$A = \frac{7+7\text{ in.}}{2} \times 3{\cdot}14 \quad = \quad 21{\cdot}98\text{ in.}$$

$$B = 2C$$

$$B = 2\,(60\text{ in.}) \quad = 120{\cdot}00\text{ in.}$$

Butt length = 141·98 in.
= 11 ft. 10 in.

(2) When ratio of pulley is 2 — 1 or more:

Add together A, B, and C.

$$A = \frac{D+d}{2} \times 3{\cdot}14\text{; e.g.}$$

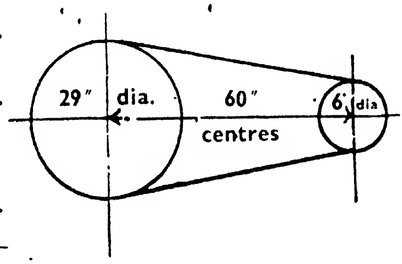
$$A = \frac{29+6\text{ in.}}{2} \times 3{\cdot}14 = 54{\cdot}95\text{ in.}$$

$$B = 2C$$

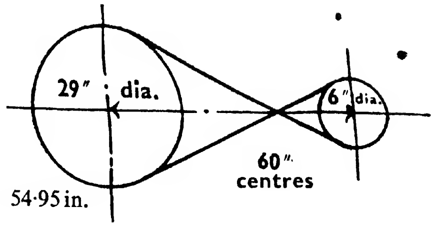
$$B = 2\,(60\text{ in.}) \quad = 120{\cdot}00\text{ in.}$$

$$C = \frac{(D-d)^2}{4C}$$

$$C = \frac{29-6\text{ in.})^2}{4\,(60\text{ in.})} \quad = \quad 2{\cdot}20\text{ in.}$$



Butt length = 177·15 in.
= 14 ft. 9⅛ in.



(3) When belt is crossed:

Add together A and B.

$$A = \frac{D + d}{2} \times 3{\cdot}14;\ \text{e.g.}$$

$$A = \frac{29 + 6 \text{ in.}}{2} \times 3{\cdot}14 \qquad = 54{\cdot}95 \text{ in.}$$

$$B = 2\sqrt{\left(\frac{D + d}{2}\right)^2 + C^2}$$

$$B = 2\sqrt{\left(\frac{29 + 6 \text{ in.}}{2}\right)^2 + 60^2} = 125{\cdot}00 \text{ in.}$$

Butt length $= 179{\cdot}95$ in.
$= 15$ ft.

To calculate the changed length of a belt when a different size of pulley is put on in place of one removed: add to, or subtract from, the present length of belt, one and a half times the difference in diameters of the old and new pulleys, e.g. take off 10-in. pulley and put on a 16-in. pulley; difference in diameters is 6 in. Multiply $6 \times 1\frac{1}{2} = 9$ in. of new belting to be added to existing one.

To find diameter of driven pulley:

$$\text{Diameter} = \frac{\text{Diameter of driver} \times \text{R.P.M. of driver}}{\text{R.P.M. of driven}}$$

$$\text{e.g.} \quad \frac{10 \text{ in.} \times 250 \text{ R.P.M.}}{125 \text{ R.P.M.}} = 20 \text{ in. diameter.}$$

To find R.P.M. of driven pulley:

$$\text{R.P.M.} = \frac{\text{Diameter of driver} \times \text{R.P.M. of driver}}{\text{Diameter of driven}}$$

$$\text{e.g.} \quad \frac{10 \text{ in.} \times 250 \text{ R.P.M.}}{20 \text{ in.}} = 125 \text{ R.P.M.}$$

To find belt speed in feet per minute (F.P.M.):

$$S = \frac{\text{Diameter in inches} \times 3{\cdot}14 \times \text{R.P.M.}}{12}$$

$$\text{e.g.} \quad \frac{10 \text{ in.} \times 3{\cdot}14 \times 250}{12} = 654 \text{ F.P.M.}$$

Effective tension of leather belting:

The pull per inch of width with 180° arc of contact is:

Single-ply . . . . 50 lb. per inch of width.
Double-ply . . . . 80 lb. per inch of width.

Three-ply is capable of developing and withstanding an effective tension equal to twice that of a single belt.

Effective tension must be corrected when the arc of contact on the smaller pulley is less than 180°, or if thickness of belt is less than standard.

To find the arc of contact on the smaller pulley when the drive is open and without idler or jockey pulleys:

$$\text{Arc of contact} = 180^\circ - \frac{60\,(D - d)}{\text{Centres in inches}}$$

$$\text{e.g. } 180^\circ - \frac{60\,(25\text{ in.} - 5\text{ in.})}{30\text{-in. centres}} = 140^\circ$$

A table giving the arc of contact correction factors for V-belts will be found on page 480.

To find horse-power belt will transmit:

$$\text{H.P.} = \frac{\text{C.E.T. in lb. per inch} \times W \times S}{33000}$$

(Example: 3 in. × 4 mm. single belt at 1,000 F.P.M.)

$$= \frac{45 \times 3 \times 1000}{33000} = 4{\cdot}1 \text{ H.P. belt will transmit.}$$

To find width of belt required:

$$W = \frac{\text{H.P.} \times 33000}{\text{C.E.T. in lb. per inch} \times S}$$

(Example : H.P. required, 5; belt speed, 1,000 R.P.M.)

$$= \frac{4 \times 33000}{45 \times 1000} = 3 \text{ in. Say 3-in. single belt.}$$

It should be remembered, however, that the examples only take into account the correction for effective tension, and the answers must therefore be considered theoretical, being based on engineering formulæ. In practice many other factors must be considered before arriving at the horse-power which a belt will safely transmit, such as atmospheric conditions and the nature of the load.

FIG. 8.—SECTION OF V-TYPE BELT

This example is made with cotton duck impregnated with balata gum, similar to balata belting.

FIG. 9.—A DRIVE UTILISING SIX V-BELTS TO A LOOM IN A LANCASHIRE COTTON MILL (*J. H. Fenner & Co., Ltd.*)

Minimum belt speed should be not less than 200 ft. per minute. Maximum belt speed should be not more than 5,000/6,000 ft. per minute. Over 4,800 ft. per minute the centrifugal force of the belt has a tendency to lift the belt from the face of the pulley, thereby reducing the arc of contact and the horse-power transmitted.

PULLEY ALIGNMENT.—Belting cannot give good results if the pulley or shafting is out of alignment. Alignment should be checked once per year, as the shifting of heavy loads on floors above may be the cause of distortion.

If the drive has a heavy starting torque or fluctuating load, it is always advisable to belt for the maximum horse-power. Let the belts be on top of their work in preference to working to their overload capacity. A belt having an inch or two to spare over the rated width will amply repay the slight extra initial cost.

V-BELTS

These belts may be composed of layers of leather or alternative layers of fabric and rubber with balata covering. With the exception of the leather variety, all belts are moulded in one piece to the required specifications, and therefore do not have to be joined. V-belts have also been made with steel cables sandwiched between two layers of rubber. Special belts are also manufactured having heat- and oil-resisting properties. For V-belt drives the ordinary

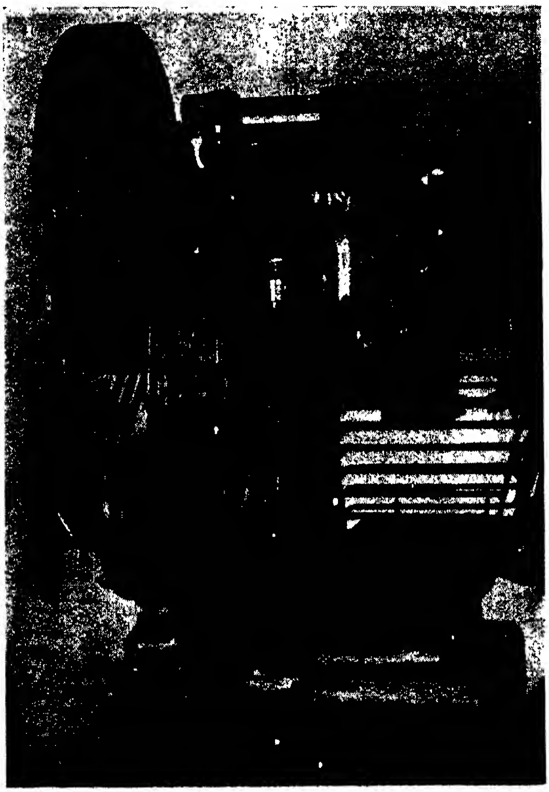


FIG. 10.—A DRIVE TO A 10-TON OVERHANG CRANK PRESS USING THREE V-BELTS

The belts transmit power from a 1-h.p. electric motor at 940 r.p.m. (*Humphris & Sons, Ltd.*)

flat pulley is usually replaced by single multi-grooved pulleys to allow the belt to have a wedge-grip inside. However, on drives with a speed ratio of 3 to 1 or more, it is possible to replace the grooved pulley by a flat pulley. The frictional grip provided by the base of the V-belts on the flat pulley will not then be less than the wedge-grip in a grooved pulley. The grooved pulleys are mostly made of cast iron, and it should be understood that the grooves must be accurately turned and the pulley carefully balanced if trouble is to be avoided.

For best life and service, pulleys larger than the maker's recommended minimum should be used, and the centre distance should be as long as reasonably convenient. A drive working with minimum diameter pulleys and shortest centre distance cannot be expected to give as long a life as one using larger pulleys and working at long centres. In these cases the number of belts should be increased, so as to counteract the extra wear and tear. The normal minimum centre distance should be not more than the diameter of the larger pulley, and for shorter centres than this 20–30 per cent. more belts should be used. A useful criterion for wear and tear is the number of trips the belts make in space per second, five or six being a normal figure.

Minimum belt speed should be not less than 200 ft. per minute. Maximum belt speed should be not more than 5,000/6,000 ft. per minute. Over 4,800 ft. per minute the centrifugal force of the belt has a tendency to lift the belt from the face of the pulley, thereby reducing the arc of contact and the horse-power transmitted.

PULLEY ALIGNMENT.—Belting cannot give good results if the pulley or shafting is out of alignment. Alignment should be checked once per year, as the shifting of heavy loads on floors above may be the cause of distortion.

FIG. 9.—A DRIVE UTILISING SIX V-BELTS TO A LOOM IN A LANCASHIRE COTTON MILL (*J. H. Fenner & Co., Ltd.*)

If the drive has a heavy starting torque or fluctuating load, it is always advisable to belt for the maximum horse-power. Let the belts be on top of their work in preference to working to their overload capacity. A belt having an inch or two to spare over the rated width will amply repay the slight extra initial cost.

V-BELTS

These belts may be composed of layers of leather or alternative layers of fabric and rubber with balata covering. With the exception of the leather variety, all belts are moulded in one piece to the required specifications, and therefore do not have to be joined. V-belts have also been made with steel cables sandwiched between two layers of rubber. Special belts are also manufactured having heat- and oil-resisting properties. For V-belt drives the ordinary

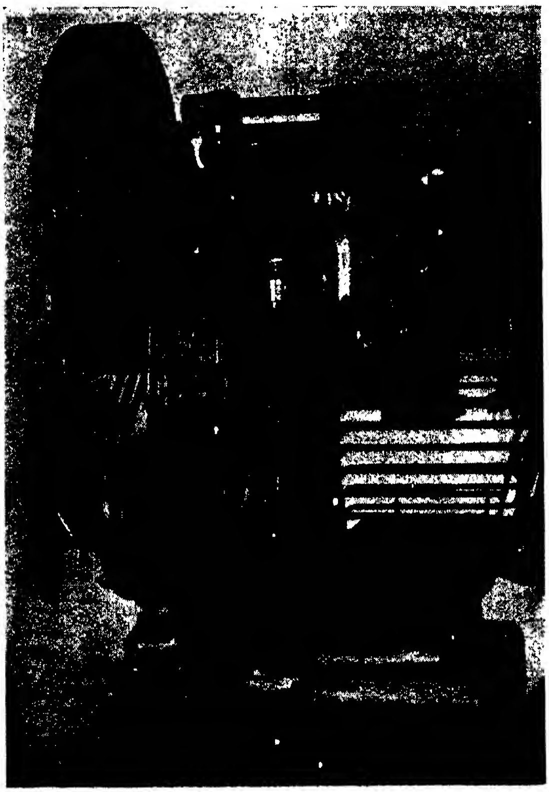


FIG. 10.—A DRIVE TO A 10-TON OVERHANG CRANK PRESS USING THREE V-BELTS

The belts transmit power from a 1-h.p. electric motor at 940 r.p.m. (*Humphris & Sons, Ltd.*)

flat pulley is usually replaced by single multi-grooved pulleys to allow the belt to have a wedge-gripinside. However, on drives with a speed ratio of 3 to 1 or more, it is possible to replace the grooved pulley by a flat pulley. The frictional grip provided by the base of the V-belts on the flat pulley will not then be less than the wedge-grip in a grooved pulley. The grooved pulleys are mostly made of cast iron, and it should be understood that the grooves must be accurately turned and the pulley carefully balanced if trouble is to be avoided.

For best life and service, pulleys larger than the maker's recommended minimum should be used, and the centre distance should be as long as reasonably convenient. A drive working with minimum diameter pulleys and shortest centre distance cannot be expected to give as long a life as one using larger pulleys and working at long centres. In these cases the number of belts should be increased, so as to counteract the extra wear and tear. The normal minimum centre distance should be not more than the diameter of the larger pulley, and for shorter centres than this 20–30 per cent. more belts should be used. A useful criterion for wear and tear is the number of trips the belts make in space per second, five or six being a normal figure.

CORRECTION OF FAULTS IN V-BELT DRIVES

ABRASION OF BELT.—Metal particles, sawdust, and grit are the usual causes, and entry of these may be stopped by covering the top and/or side of the drive.

CRACKS ON SIDE AND BASE.—These are generally caused by excess heat and chemical fumes, but can be prevented by installing special heat-resisting belts. In any case, V-belts should not be installed in a hot place, such as near a radiator or on drying machines and kilns.

EXCESSIVE STRETCH.—This is usually due to overloading or internal breaks which are caused by forcing the belt into the pulley groove.

Extra duty may have been added to the drive design and installation.

LOSS IN DRIVEN SPEED.—This is often due to slippage resulting from insufficient tension. Check by tachometer readings against calculated speed and adjust tension. Slippage may result from the use of incorrect belt dressings.

MIXING OF BELTS.—Belts of different brands or types must not be mixed. Differences, however slight, in cross section and internal design will seriously affect running life.

New and used belts should not be run together for similar reasons.

When replacing belts, all new or all used belts should run in the same set.

INABILITY TO TRANSMIT RATED LOAD.– This may be the result of the pulley grooves being cut unevenly, or of incorrect angle. Check each single groove with a template to ensure all belts are running on the same pitch diameter.

Have the drive design checked to see that the drive has sufficient capacity for the load.

DRIVE DESIGN FORMULÆ FOR V-BELTS

The symbols used in the formulæ are:

H.P. = Horse-power.
R.P.M. = Revolutions per minute.
R = Speed ratio.
C = Centre distance in inches.
L = Outside length of V-belt in inches.
$p.d.$ = Pitch diameter of pulley.
d = Outside diameter of small pulley in inches.
D = Outside diameter of large pulley in inches.
V = V-belt velocity in feet per minute.
F = Arc of contact correction factor (Table VI).
K = Service correction factor.

1. *Speed Ratio.* $R = \dfrac{\text{R.P.M. of high-speed shaft}}{\text{R.P.M. of low-speed shaft}}$

2. *V-belt Velocity.* $V = 0{\cdot}262 \times p.d. \times \text{R.P.M. of } p.d.$

3. *V-belt Length (Outside).* $L = 2C + \dfrac{(D-d)^2}{4C} + 1{\cdot}57(D + d)$

4. *Centre Distance.* $C = A + \sqrt{A^2 - B}$

$$\text{where } A = \frac{L}{4} - 0{\cdot}3925(D + d)$$

$$B = \frac{(D - d)^2}{8}$$

5. *Number of V-belts required.*

$$= \frac{\text{Total H.P. to be transmitted}}{\text{Power for V-belt at given velocity} \times F \times K}$$

V-belt lengths and centre distances (Formulæ 3 and 4) may be calculated also from pitch diameters and pitch lengths, and if this method is used, D is the *pitch* diameter of the large pulley, d the *pitch* diameter of the smaller pulley, and L the *pitch* length of the V-belt. In the case of Formula 3, the appropriate difference between pitch and inside lengths must be deducted instead of the difference between outside and inside lengths in order to arrive at the belt symbol number in the standard tables.

METHOD OF USING V-BELT DRIVE DESIGN FORMULÆ FOR THE CALCULATION OF ANY DRIVE FROM BASIC PRINCIPLES

The following example illustrates the method of using the drive-design formula.

To Design a Drive

From a 15-h.p. oil engine at 600 r.p.m. to a centrifugal pump to run at 2,500 r.p.m. on a centre distance of approximately 36 in.:

(*a*) SELECTION OF V-BELT SECTION

Normally "*C*" section V-belts are used for transmitting 15 h.p., but as this is a speed increase drive, and the smallest recommended pulley with "*C*" section V-belts is 8¾ in. o.d., the V-belt velocity would be 5,725 f.p.m. when running at 2,500 r.p.m. This speed is excessive, and it is therefore desirable to employ smaller-section V-belts and corresponding pulleys so that the V-belt speed does not exceed the recommended maximum of 4,500 f.p.m. "*B*" section V-belts should thus be used for this high-speed drive.

(*b*) SELECTION OF PULLEY DIAMETERS

This is a speed-increasing drive, and the larger pulley is the motor pulley, i.e. the driving pulley.

A standard-diameter driving pulley to give a speed in the region of 4,000 f.p.m. should first be selected by calculation, and for this purpose the following formula is used:

Let X equal the required pulley diameter in inches.

CORRECTION OF FAULTS IN V-BELT DRIVES

ABRASION OF BELT.—Metal particles, sawdust, and grit are the usual causes, and entry of these may be stopped by covering the top and/or side of the drive.

CRACKS ON SIDE AND BASE.—These are generally caused by excess heat and chemical fumes, but can be prevented by installing special heat-resisting belts. In any case, V-belts should not be installed in a hot place, such as near a radiator or on drying machines and kilns.

EXCESSIVE STRETCH.—This is usually due to overloading or internal breaks which are caused by forcing the belt into the pulley groove.

Extra duty may have been added to the drive design and installation.

LOSS IN DRIVEN SPEED.—This is often due to slippage resulting from insufficient tension. Check by tachometer readings against calculated speed and adjust tension. Slippage may result from the use of incorrect belt dressings.

MIXING OF BELTS.—Belts of different brands or types must not be mixed. Differences, however slight, in cross section and internal design will seriously affect running life.

New and used belts should not be run together for similar reasons.

When replacing belts, all new or all used belts should run in the same set.

INABILITY TO TRANSMIT RATED LOAD.—This may be the result of the pulley grooves being cut unevenly, or of incorrect angle. Check each single groove with a template to ensure all belts are running on the same pitch diameter.

Have the drive design checked to see that the drive has sufficient capacity for the load.

DRIVE DESIGN FORMULÆ FOR V-BELTS

The symbols used in the formulæ are:

H.P. = Horse-power.
R.P.M. = Revolutions per minute.
R = Speed ratio.
C = Centre distance in inches.
L = Outside length of V-belt in inches.
$p.d.$ = Pitch diameter of pulley.
d = Outside diameter of small pulley in inches.
D = Outside diameter of large pulley in inches.
V = V-belt velocity in feet per minute.
F = Arc of contact correction factor (Table VI).
K = Service correction factor.

1. *Speed Ratio.* $R = \dfrac{\text{R.P.M. of high-speed shaft}}{\text{R.P.M. of low-speed shaft}}$

2. *V-belt Velocity.* $V = 0{\cdot}262 \times p.d. \times \text{R.P.M. of } p.d.$

3. *V-belt Length (Outside).* $L = 2C + \dfrac{(D - d)^2}{4C} + 1{\cdot}57(D + d)$

4. *Centre Distance.* $C = A + \sqrt{A^2 - B}$

$$\text{where } A = \frac{L}{4} - 0{\cdot}3925(D + d)$$

$$B = \frac{(D - d)^2}{8}$$

5. *Number of V-belts required.*

$$= \frac{\text{Total H.P. to be transmitted}}{\text{Power for V-belt at given velocity} \times F \times K}$$

V-belt lengths and centre distances (Formulæ 3 and 4) may be calculated also from pitch diameters and pitch lengths, and if this method is used, D is the *pitch* diameter of the large pulley, d the *pitch* diameter of the smaller pulley, and L the *pitch* length of the V-belt. In the case of Formula 3, the appropriate difference between pitch and inside lengths must be deducted instead of the difference between outside and inside lengths in order to arrive at the belt symbol number in the standard tables.

METHOD OF USING V-BELT DRIVE DESIGN FORMULÆ FOR THE CALCULATION OF ANY DRIVE FROM BASIC PRINCIPLES

The following example illustrates the method of using the drive-design formula.

To Design a Drive

From a 15-h.p. oil engine at 600 r.p.m. to a centrifugal pump to run at 2,500 r.p.m. on a centre distance of approximately 36 in.:

(*a*) SELECTION OF V-BELT SECTION

Normally "*C*" section V-belts are used for transmitting 15 h.p., but as this is a speed increase drive, and the smallest recommended pulley with "*C*" section V-belts is 8¾ in. o.d., the V-belt velocity would be 5,725 f.p.m. when running at 2,500 r.p.m. This speed is excessive, and it is therefore desirable to employ smaller-section V-belts and corresponding pulleys so that the V-belt speed does not exceed the recommended maximum of 4,500 f.p.m. "*B*" section V-belts should thus be used for this high-speed drive.

(*b*) SELECTION OF PULLEY DIAMETERS

This is a speed-increasing drive, and the larger pulley is the motor pulley, i.e. the driving pulley.

A standard-diameter driving pulley to give a speed in the region of 4,000 f.p.m. should first be selected by calculation, and for this purpose the following formula is used:

Let X equal the required pulley diameter in inches.

HORSE-POWER RATING TABLES

TABLE I.—*A*-SECTION[1] V-BELTS

(With 180° Arc of Contact on smaller Pulley)

V-belt Velocity (ft. per min.)	*Horse-power Ratings*					
	Pulley Pitch Diameter 2·6 in.	*Pulley Pitch Diameter 3 in.*	*Pulley Pitch Diameter 3·4 in.*	*Pulley Pitch Diameter 3·8 in.*	*Pulley Pitch Diameter 4·2 in.*	*Pulley Pitch Diameter 5 in. and larger*
500	0·3	0·3	0·4	0·5	0·5	0·5
600	0·3	0·4	0·5	0·6	0·6	0·6
700	0·4	0·5	0·6	0·6	0·6	0·7
800	0·4	0·6	0·7	0·7	0·7	0·8
900	0·5	0·6	0·7	0·8	0·8	0·9
1,000	0·5	0·7	0·8	0·9	0·9	1·0
1,200	0·6	0·8	1·0	1·0	1·1	1·2
1,400	0·7	0·9	1·1	1·2	1·3	1·4
1,600	0·8	1·0	1·2	1·3	1·4	1·6
1,800	0·9	1·2	1·4	1·5	1·6	1·8
2,000	1·0	1·3	1·5	1·6	1·8	2·0
2,200	1·1	1·4	1·6	1·8	1·9	2·2
2,400	1·1	1·4	1·7	2·0	2·0	2·3
2,600	—	1·6	1·8	2·0	2·2	2·5
2,800	—	1·6	1·9	2·1	2·3	2·6
3,000	—	—	2·0	2·2	2·4	2·7
3,500	—	—	—	2·5	2·7	3·0
4,000	—	—	—	—	2·8	3·3
4,500	—	—	—	—	—	3·4

[1] *Details of the standard sections, A, B, C, D, and E, will be found on page* 471.

TABLE II.—*B*-SECTION V-BELTS

(With 180° Arc of Contact on Smaller Pulley)

V-belt Velocity (ft. per min.)	*Horse-power Ratings*				
	Pulley Pitch Diameter 5 in.	*Pulley Pitch Diameter 5·4 in.*	*Pulley Pitch Diameter 5·8 in.*	*Pulley Pitch Diameter 6·2 in.*	*Pulley Pitch Diameter 7 in. and larger*
500	0·7	0·8	0·8	0·9	1·0
600	0·9	0·9	1·0	1·1	1·2
700	1·0	1·0	1·1	1·2	1·3
800	1·1	1·2	1·2	1·3	1·5
900	1·2	1·3	1·4	1·5	1·6
1,000	1·3	1·4	1·5	1·6	1·8
1,200	1·5	1·6	1·7	1·8	2·0
1,400	1·7	1·9	2·0	2·1	2·3
1,600	1·9	2·1	2·3	2·4	2·6
1,800	2·2	2·4	2·5	2·7	2·9
2,000	2·4	2·6	2·8	2·9	3·2
2,200	2·5	2·8	3·0	3·2	3·5
2,400	2·7	3·0	3·2	3·4	3·7
2,600	2·9	3·2	3·4	3·6	4·0
2,800	3·0	3·3	3·6	3·8	4·3
3,000	3·2	3·5	3·8	4·0	4·5
3,500	—	3·8	4·1	4·4	4·9
4,000	—	—	4·4	4·7	5·3
4,500	—	—	—	4·8	5·4
5,000	—	—	—	—	5·5

TABLE III.—*C*-SECTION V-BELTS

(With 180° Arc of Contact on Smaller Pulley)

V-belt Velocity (ft. per min.)	*Horse-power Ratings*				
	Pulley Pitch Diameter 7 in.	*Pulley Pitch Diameter 8 in.*	*Pulley Pitch Diameter 9 in.*	*Pulley Pitch Diameter 10 in.*	*Pulley Pitch Diameter 12 in. and larger*
500	1·1	1·4	1·5	1·6	1·9
600	1·3	1·6	1·7	1·9	2·3
700	1·5	1·8	2·0	2·3	2·7
800	1·6	2·1	2·3	2·6	3·0
900	1·8	2·3	2·6	2·9	3·3
1,000	2·0	2·5	2·8	3·1	3·6
1,200	2·4	2·9	3·4	3·8	4·3
1,400	2·7	3·4	3·9	4·4	5·0
1,600	3.1	3·9	4·5	4·9	5·7
1,800	3·4	4·3	5·0	5·5	6·3
2,000	3·7	4·7	5·5	6·1	7·0
2,200	4·0	5·1	5·9	6·6	7·6
2,400	4·3	5·5	6·4	7·1	8·2
2,600	4·6	5·8	6·8	7·6	8·8
2,800	4·8	6·2	7·2	8·0	9·3
3,000	5·0	6·5	7·6	8·5	9·8
3,500	—	7·1	8·4	9·4	11·0
4,000	—	—	9·0	10·2	12·0
4,500	—	—	—	10·7	12·7
5,000	—	—	—	—	13·2

TABLE IV.—*D*-SECTION V-BELTS

(With 180° Arc of Contact on Smaller Pulley)

V-belt Velocity (ft. per min.)	*Horse-power Ratings*				
	Pulley Pitch Diameter 12 in.	*Pulley Pitch Diameter 13 in.*	*Pulley Pitch Diameter 14 in.*	*Pulley Pitch Diameter 15 in.*	*Pulley Pitch Diameter 17 in. and larger*
500	2·3	2·6	3·0	3·5	3·8
600	2·7	3·2	3·6	4·0	4·4
700	3·2	3·6	4·2	4·6	4·9
800	3·6	4·2	4·7	5·1	5·6
900	4·0	4·7	5·1	5·6	6·2
1,000	4·5	5·1	5·6	6·1	6·8
1,200	5·3	6·1	6·7	7·3	8·2
1,400	6.1	7·0	7·8	8·4	9·4
1,600	6·9	7·9	8·8	9·5	10·7
1,800	7·7	8·8	9·8	10·6	12·0
2,000	8·4	9·7	10·7	11·7	13·2
2,200	9·1	10·5	11·7	12·7	14·4
2,400	9·8	11·3	12·6	13·7	15·5
2,600	10·4	12·0	13·4	14·6	16·6
2,800	10·9	12·7	14·1	15·5	17·6
3,000	11·4	13·3	14·9	16·3	18·6
3,500	12·4	14·6	16·5	18·1	20·7
4,000	—	15·5	17·6	19·4	22·5
4,500	—	—	—	20·3	23·5
5,000	—	—	—	—	24·4

TABLE V.—*E*-SECTION V-BELTS
(With 180° Arc of Contact on Smaller Pulley)

V-belt Velocity (ft. per min.)	*Horse-power Ratings*				
	Pulley Pitch Diameter 20 *in.*	*Pulley Pitch Diameter* 22 *in.*	*Pulley Pitch Diameter* 24 *in.*	*Pulley Pitch Diameter* 26 *in.*	*Pulley Pitch Diameter* 28 *in. and larger*
500	3·6	4·3	4·9	5·5	6·0
600	4·5	5·3	6·0	6·6	7·0
700	5·4	6·3	6·9	7·6	8·1
800	6·4	7·2	8·0	8·7	9·2
900	7·2	8·1	8·9	9·6	10·2
1,000	8·0	9·0	9·8	10·5	11·1
1,200	9·5	10·7	11·7	12·6	13·3
1,400	11·0	12·4	13·6	14·6	15·4
1,600	12·4	14·1	15·4	16·6	17·5
1,800	13·8	15·7	17·2	18·5	19·6
2,000	15·2	17·2	18·9	20·3	21·6
2,200	16·5	18·7	20·6	22·1	23·5
2,400	17·7	20·1	22·2	23·9	25·3
2,600	18·9	21·5	23·7	25·5	27·1
2,800	19·9	22·8	25·1	27·1	29·9
3,000	20·9	24·0	26·5	28·6	30·5
3,500	23·0	26·6	29·5	32·0	34·2
4,000	24·5	28·5	31·9	34·7	37·2
4,500	—	—	33·5	36·7	39·5
5,000	—	—	—	37·9	40·9

METHOD OF USING ARC OF CONTACT CORRECTION FACTOR

To determine correction factor, subtract the outside diameter of the small pulley from the outside diameter of the large pulley. In the column headed "Difference in Pulley Diameters," select the figure nearest the difference obtained, and read across from the centre distance line to this column. The correction factor is given at the point of intersection (see Table IV, page 480).

EXAMPLE:

Outside diameter of driving pulley . . . . = 10 in.
Outside diameter of driven pulley . . . = 30 in.
Centre distance . . . . = 40 in.
Difference in pulley diameters . . . . = 30 in. — 10 in.
= 20 in.

Under the heading, "Difference in Pulley Diameters," find the column "20" and read down to the point where this column intersects the 40-in. centre distance line. The correction factor is found by reading *across* the 40-in. centre distance line, and *down* the 20-in. difference in pulley diameters column, and in this example will be found to be 0·92.

Interpolation is permissible; for example, where the difference in pulley diameters is 11 in. and the centre distance is 20 in., the arc of contact service correction factor is 0·91. Similarly, where the difference in pulley diameters is 14 in. and the centre distance 45 in., the service correction factor is 0·96.

TABLE VI.—ARC OF CONTACT FACTORS (*F*) FOR CORRECTING HORSE-POWER RATINGS
(Multiply basic h.p. at 180° by appropriate factor if pulley diameters differ.)

Centre Distance in in.	*Difference in Pulley Diameters—in.*																				
	2	4	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	42	48	54	60
10	0·98	0·94	0·91	0·86	0·83	0·78	0·71															
15	0·98	0·96	0·93	0·91	0·89	0·86	0·84	0·81	0·78	0·74	0·69											
20	0·99	0·97	0·95	0·94	9·92	0·90	0·89	0·87	0·85	0·83	0·80	0·78	0·74	0·72	0·69							
25	0·99	0·98	0·97	0·95	0·94	0·92	0·91	0·89	0·87	0·86	0·85	0·84	0·82	0·80	0·78	0·76	0·73	0·71				
30	0·99	0·98	0·98	0·97	0·95	0·94	0·93	0·91	0·90	0·89	0·88	0·87	0·85	0·84	0·83	0·82	0·80	0·78	0·72			
40	0·99	0·99	0·98	0·97	0·97	0·95	0·95	0·93	0·93	0·92	0·91	0·90	0·89	0·88	0·87	0·87	0·86	0·85	0·82	0·78	0·74	0·69
50	1·00	0·99	0·99	0·98	0·98	0·97	0·97	0·95	0·95	0·94	0·93	0·92	0·92	0·91	0·90	0·89	0·89	0·88	0·86	0·84	0·81	0·78
60	1·00	0·99	0·99	0·98	0·98	0·98	0·97	0·96	0·95	0·95	0·94	0·93	0·93	0·93	0·92	0·91	0·91	0·90	0·88	0·87	0·85	0·83
70	1·00	0·99	0·99	0·99	0·98	0·98	0·98	0·97	0·96	0·96	0·95	0·95	0·94	0·94	0·93	0·93	0·92	0·91	0·90	0·89	0·87	0·86
80	1·00	0·99	0·99	0·99	0·98	0·98	0·98	0·98	0·97	0·96	0·96	0·96	0·95	0·95	0·94	0·94	0·93	0·93	0·91	0·90	0·89	0·88
90	1·00	1·00	0·99	0·99	0·99	0·98	0·98	0·98	0·98	0·97	0·97	0·96	0·96	0·95	0·95	0·95	0·94	0·94	0·93	0·91	0·90	0·89
100	1·00	1·00	0·99	0·99	0·99	0·99	0·98	0·98	0·98	0·97	0·97	0·97	0·96	0·96	0·96	0·96	0·95	0·95	0·93	0·93	0·91	0·90

H.P. RATINGS, LOW-SPEED DRIVES

Where it is necessary to design an ultra-low-speed drive with a V-belt speed under 500 f.p.m., it is always desirable to check the actual horse-power and to see that the shaft is *capable* of transmitting this figure.

The following table includes the horse-power which medium steel shafting is capable of transmitting at low speeds and is subject to the following corrections:

For main transmitting shafts, multiply ratings below by 1·0.
For line shafts carrying pulleys, multiply ratings below by 1·5.
For short shafts, multiply ratings below by 2·0.

TABLE VII.—H.P. OF MEDIUM STEEL SHAFTING AT LOW SPEEDS

R.P.M.	Diameter of Shaft in in.																		
	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	4
20	0·03	0·06	0·10	0·17	0·25	0·36	0·49	0·65	0·84	1·34	2·0	2·6	3·5	5·2	6·8	8·6	10·7	13·2	16·0
30	0·05	0·09	0·16	0·25	0·37	0·53	0·73	0·97	1·26	2·01	3·0	3·9	5·2	7·8	10·2	12·9	16·1	19·8	24·0
40	0·06	0·12	0·21	0·33	0·50	0·71	0·98	1·30	1·68	2·68	4·0	5·2	6·9	10·4	13·5	17·2	21·4	26·4	32·0
50	0·08	0·15	0·26	0·42	0·62	0·89	1·22	1·62	2·10	3·34	5·0	6·5	8·7	13·0	16·9	21·5	26·8	33·0	40·0
60	0·09	0·18	0·31	0·50	0·75	1·07	1·46	1·95	2·52	4·02	6·0	7·8	10·4	15·6	20·2	25·8	32·1	39·6	48·0
70	0·11	0·21	0·37	0·58	0·87	1·24	1·71	2·27	2·94	4·68	7·0	9·1	12·2	18·2	23·6	30·1	37·5	46·2	56·0
80	0·12	0·24	0·47	0·67	1·00	1·42	1·95	2·60	3·36	5·36	8·0	10·4	13·9	20·8	27·0	34·3	42·8	52·8	64·0
90	0·14	0·27	0·47	0·75	1·12	1·60	2·19	2·92	3·78	6·03	9·0	11·7	15·6	23·4	30·4	38·7	48·2	59·3	72·0
100	0·16	0·30	0·52	0·82	1·25	1·78	2·44	3·24	4·20	6·70	10·0	13·0	17·3	26·0	33·7	43·0	53·6	66·0	80·0

TABLE VIII.—RELATION BETWEEN INSIDE, PITCH, AND OUTSIDE DIAMETERS AND LENGTHS

V-belt Section	*Pulley Outside Diameter exceeds Pulley Pitch Diameter by (in.)*	*V-belt Pitch Length exceeds Inside Length by (in.)*	*V-belt Outside Length exceeds Inside Length by (in.)*
A	$\frac{3}{8}$	0·98	2·16
B	$\frac{1}{2}$	1·18	2·75
C	$\frac{3}{4}$	1·45	3·34
D	$\frac{7}{8}$	1·96	4·71
E	$1\frac{1}{8}$	2·75	6·28

TABLE IX.—OUTSIDE PULLEY DIAMETERS FOR STANDARD V-BELTS

V-belt Section	*Recommended Minimum (in in.)*	*Absolute Minimum (in in.)*
A	$3\frac{3}{8}$	$2\frac{1}{2}$
B	$5\frac{1}{2}$	4
C	$8\frac{3}{4}$	7
D	12	$9\frac{3}{4}$
E	20	17

TABLE X.—SERVICE CORRECTION FACTORS (K)

(To be used in conjunction with drive design formula 5, page 474.)

	Running hours per day		
	Up to 3	8–10	24
Steady loads	1·25	1·0	0·80
Pulsating and uneven loads . .	1·0	0·80	0·70
Heavy loads and high starting torques	0·70	0·60	0·50

MODERN REFRIGERATION PRACTICE

The following survey of modern refrigerating practice is based upon a Thomas Hawksley Lecture read before the Institution of Mechanical Engineers by Lord Dudley Gordon, D.S.O., M.I.Mech.E.

REFRIGERATION is concerned with temperatures below the normal atmospheric temperature. The lowest limit is obviously absolute zero, $-273{\cdot}144°$ C., to which the nearest approach has been made at the Kamerlingh Onnes Laboratory at Leiden, where an absolute temperature of $0{\cdot}005°$ K. has been reached.

Industrial practice begins at the liquefying temperatures of certain gases at atmospheric pressure. For instance, the temperature of liquid oxygen at atmospheric pressure is $-183°$ C., nitrogen $-195{\cdot}8°$ C., and liquid air about $-192°$ C. The liquefaction of air by Linde in 1894 soon led to the development of liquefied gases used for welding and cutting, and to the process for the synthetic production of ammonia; but in these processes refrigerating machinery is only used for precooling the original mixture of gases to a temperature of about $-60°$ F., the remainder of the process being carried out by means of compression to a moderate pressure, by rectification and heat exchange between the gases handled.

The Refrigerating Cycle

The simplest type of mechanical refrigerator consists of four main units, namely:

The Compressor.
The Condenser.
The Regulator.
The Evaporator.

THE COMPRESSOR.—This draws refrigerant vapour from the evaporator, compresses it to a predetermined degree, and, at the same time, raises its temperature and forces it through an oil separator into—

THE CONDENSER.—This is a device for extracting the heat from the refrigerant vapour, and so converting it into a liquid state.

There are various types of condenser, e.g. the shell type, which is built on similar principles to the well-known steam surface condenser, or the double-pipe type, where cooling water circulates through the inner pipe and the re-

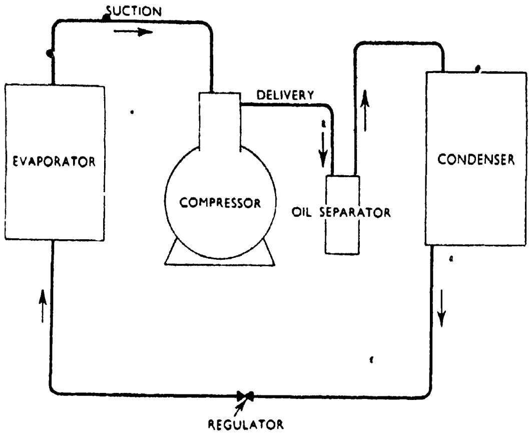


FIG. 1.—CIRCUIT FOR WET COMPRESSION WITH HAND-OPERATED REGULATOR

frigerant through the outer pipe, the refrigerant and the cooling water travelling in opposite directions.

The refrigerant now emerges from the condenser in liquid form and passes along the pipe to—

THE REGULATING VALVE.—The function of the regulating valve is to restrict the flow of the refrigerant so that the necessary pressure is maintained in the condenser to cause the cooled refrigerant vapour to liquefy. On the other side of the regulator valve a much lower pressure exists owing to the fact that it is connected through the evaporator to the suction side of the compressor.

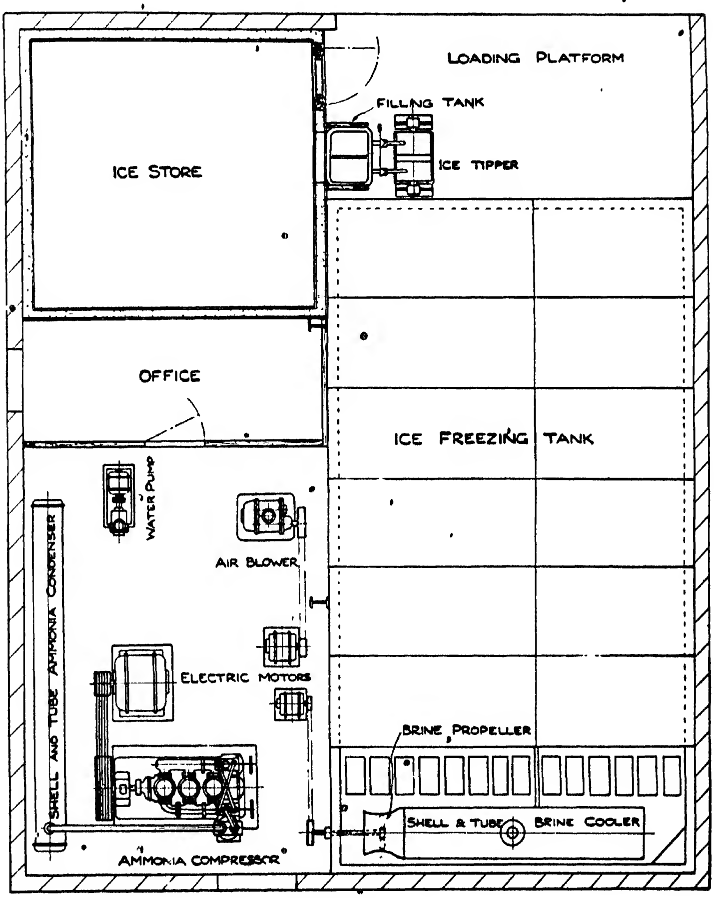
The refrigerant passes from the regulating valve into—

THE EVAPORATOR.—This is usually a series of pipes or coils into which flows all the liquid refrigerant which has not been evaporated in passing through the regulating valve. Owing to the reduced pressure, the liquid refrigerant now begins to vaporise, and, in doing so, it extracts heat first from the metal of the evaporator tubes and then from any liquid or air by which the evaporator tubes may be surrounded.

In many refrigerating systems the evaporator coils are used to cool brine, which is then circulated in the cooling chambers.

Refrigerants

For temperatures down to $-60°$ F., normal ammonia refrigerating machinery can be used economically; but there is an increasing demand in chemical and other processes for refrigerating temperatures down to $-100°$ F. or even lower, and for such temperatures refrigerants other than those in use for the usual processes to which refrigeration is applied become necessary.



PLAN

FIG. 2.—GENERAL ARRANGEMENT OF ICE-MAKING PLANT

The ice is made by freezing water contained in galvanised-iron cans which are lowered into a tank containing cold brine. The brine is usually maintained at a temperature of about 16° F. (−9° C.). The refrigerating plant consists of an ammonia compressor, ammonia condenser, and cooling coils or tubular brine cooler immersed in the ice tank. The ammonia is circulated over and over again in a closed system, and additional ammonia is required only to make up for that lost by leakage, which in a carefully operated plant is only a very small quantity.

The brine is circulated through the tubes of a shell and tube cooler, by means of a propeller which also serves to induce a circulation of the brine past the ice cans. When the cans have been frozen, they are withdrawn from the tank and placed in a tank of warm water to cause some of the ice round the edge of the block to thaw and thus free the ice block from the can. (In hot climates this tank is often dispensed with.) The ice is then tipped out of the can in the form of a block, after which the can is refilled with water and is replaced in the ice tank.

(L. Sterne & Co., Ltd.)

TABLE I.—LOW-TEMPERATURE REFRIGERANTS

Refrigerant	*Ethane,* C_2H_6	*Ethylene,* C_2H_4	*Propane,* C_3H_8
Critical point, ° F.	89·8	49·10	204·1
Triple point, ° F.	−298·5	−272·92	−309·8
Boiling-point at atmospheric pressure, ° F.	−127·5	−155·02	−48·0
Vapour pressure at 5° F., lb. per sq. in. abs.	236·5	416·40	42·1
Vapour pressure at 86° F., lb. per sq. in. abs.	682·7	—	155·3
Volume, cub. ft. per lb. at 5° F.	0·273	0·273	2·48
Volume, cub. ft. per lb. at 86° F.	0·122	—	0·717

Table I shows the triple point and the normal boiling-point of various refrigerants which may be used for these very low temperatures. Some of these are not readily obtainable. It will be seen that both ethane and ethylene are particularly suitable, and have been successfully used in large-scale applications of low temperatures. Such low temperatures are reached by means of the "cascade" system (see Fig. 14).

During the period when mechanical refrigeration was being developed to its present stage, the main uses, such as ice-making, cold storage both on board ship and on shore as well as applications in breweries, demanded temperatures from 0° F. up to about 40° F.; and at these temperatures the most suitable refrigerants were found to be ammonia, carbon dioxide, and sulphur dioxide. Table II shows the triple point and the normal boiling-point of these refrigerants.

TABLE II.—REFRIGERANTS FOR USE IN LARGE COLD STORES

Refrigerant	*Ammonia,* NH_3	*Carbon dioxide,* CO_2	*Sulphur dioxide,* SO_2
Critical point, ° F.	271·4	87·7	315·1
Triple point, ° F.	−107·86	−69·9	−98·9
Boiling-point at atmospheric pressure, ° F.	−28·0	−109·3 (solid)	13·99
Vapour pressure at 5° F., lb. per sq. in. abs.	34·27	331·9	11·71
Vapour pressure at 86° F., lb. per sq. in. abs.	169·2	1,043·0	66·85
Volume, cub. ft. per lb. at 5° F.	8·150	0·2660	6·500
Volume, cub. ft. per lb. at 86° F.	1·772	0·04789	1·265

Sulphur dioxide has several rather serious drawbacks, and to-day most ice-making plants for land use employ ammonia as the refrigerant.

On board ship and in confined spaces carbon dioxide is the most suitable refrigerant for larger-size installations in the range of temperatures under consideration.

For domestic refrigerators and air-conditioning plant other refrigerants

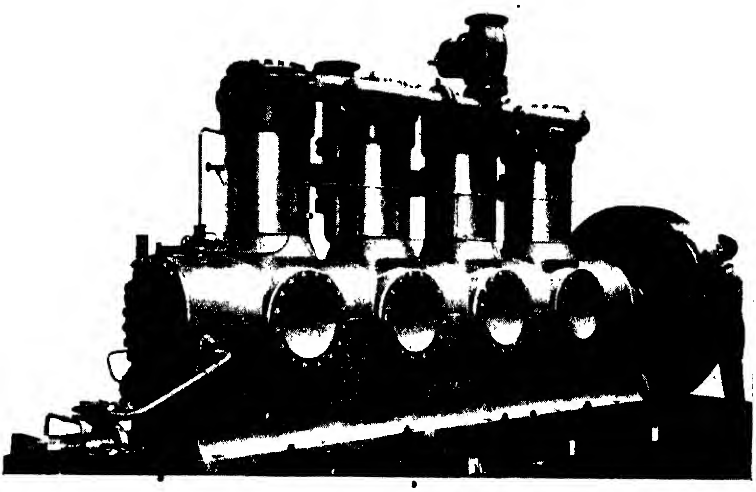


FIG. 3.—VERTICAL HIGH-SPEED AMMONIA COMPRESSOR
(*J. & E. Hall, Ltd.*)

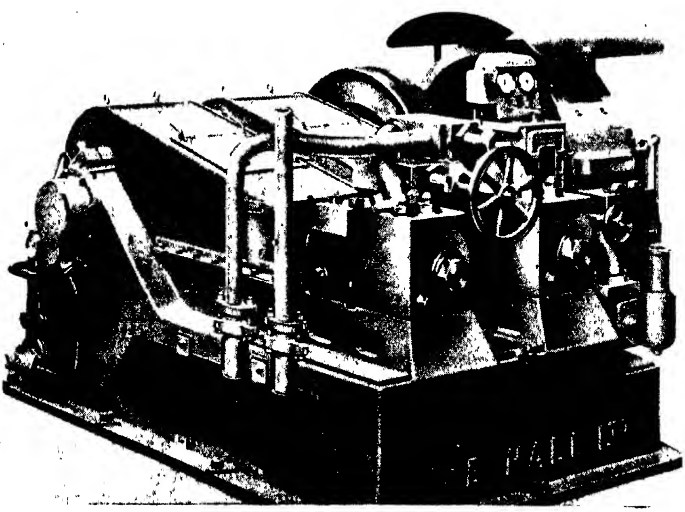


FIG. 4.—ELECTRICALLY DRIVEN HORIZONTAL CO_2 COMPRESSOR
(*J. & E. Hall, Ltd.*)

have recently come into wide use. The properties of the most important of these: methyl chloride, CH_3Cl; dichlorodifluoro-methane ("Freon" or "F 12"), CCl_2F_2, are shown in Table III.

TABLE III.—REFRIGERANTS FOR DOMESTIC AND AIR-CONDITIONING PURPOSES

Refrigerant	*Methyl chloride,* CH_3Cl	*Dichloro-difluoro-methane ("Freon" or "F 12"),* CCl_2F_2	*Methylene chloride ("Carrene 1"),* CH_2Cl_2	*Trichloro-monofluoro-methane ("Carrene 2" or "F 11"),* $CFCl_3$
Critical point, ° F.	289·6	232·7	421·0	388·4
Triple point, ° F.	−144·0	−247·0	−142·0	−168·0
Boiling-point at atmospheric pressure, ° F.	−10·76	−21·7	103·7	74·66
Vapour pressure at 5° F., lb. per sq. in. abs.	21·15	26·51	1·17	2·93
Vapour pressure at 86° F., lb. per sq. in. abs.	94·7	107·9	10·60	18·28
Volume, cub. ft. per lb. at 5° F.	4·471	1·485	50·0	12·195
Volume, cub. ft. per lb. at 86° F.	1·081	0·389	6·667	2·242

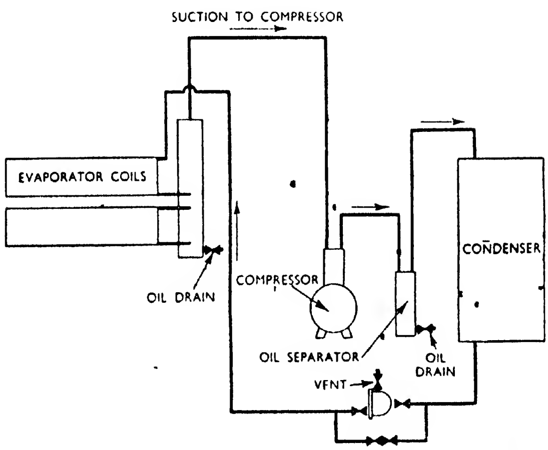


FIG. 5.—CIRCUIT FOR DRY COMPRESSION, WITH HIGH-PRESSURE FLOAT REGULATOR

Modern Refrigerating Compressors

The original refrigerating machines were in nearly every case horizontal, slow-running, double-acting compressors, the piston being designed with spherical ends and the valves so arranged in the end covers that the clearance was reduced to a minimum.

The horizontal type of compressor, particularly in the ordinary commercial large-size refrigerating plant making use of ammonia, continued in favour in Germany and most other countries on the continent of Europe up to quite recent years. During the past twenty years the high-speed vertical compressor has to an increasing extent replaced the horizontal compressor in this country

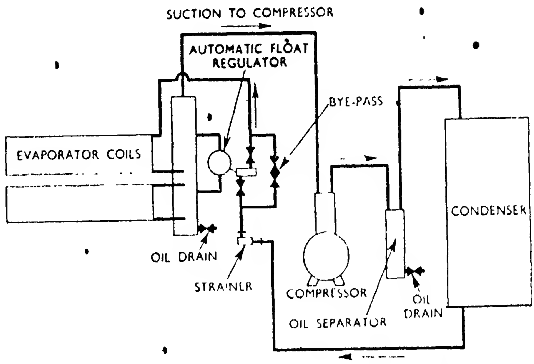


FIG. 6.—SAME CIRCUIT AS IN FIG. 5, BUT WITH LOW-PRESSURE FLOAT REGULATOR

and in the United States of America. The adoption of the vertical type has been accompanied by the usual increase in speed, which is the result of the development of modern materials and modern workshop practice. This change has had the usual effect of saving a large amount of space, and the area now required to be set aside for a large refrigerating installation is only a fraction of what it was thirty or more years ago. Fig. 3 shows a high-speed vertical compressor constructed for use with ammonia as the refrigerant.

Many different types of compressors are available to-day, including the multi-stage turbo-compressors. Most of these are of the larger sizes, and their development has depended, not only upon progress in design, but also on the use of refrigerants with a high boiling-point and a high molecular weight. Ammonia, ethyl chloride, and "Freon 11" have been used in turbo-compressors, the latter refrigerant working in most cases below atmospheric pressure. This avoids the risk of outward leakage of the refrigerant, but special precautions must be taken to prevent air from being drawn into the system. In some cases a special vacuum pump has even been employed to overcome this risk.

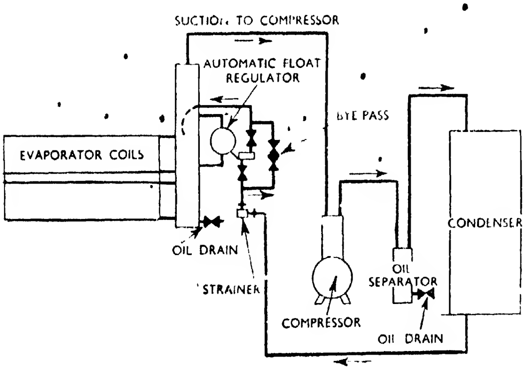


FIG. 7.—CIRCUIT FOR DRY COMPRESSION; DIRECT FEED INTO SEPARATOR

Another change which follows the development of other types of compressor is the increasing adoption of the V-belt drive, which in recent years has tended to displace the arrangement by which compressors were coupled directly to the electric motor or other prime

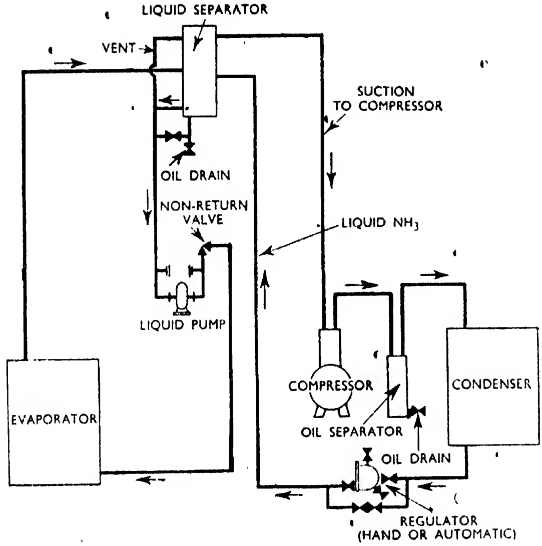


FIG. 8.—CIRCUIT COMPRISING PUMP FOR CIRCULATION OF EVAPORATING LIQUID: HIGH-PRESSURE FLOAT REGULATOR

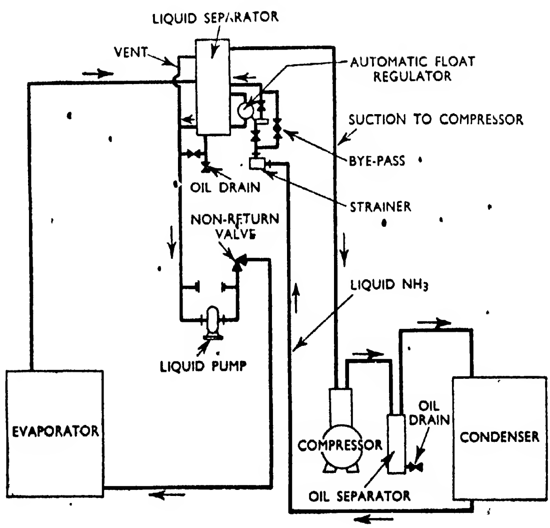


FIG. 9.—CIRCUIT COMPRISING PUMP FOR CIRCULATION OF EVAPORATING LIQUID: LOW-PRESSURE REGULATOR

mover, though the latter arrangement is still favoured for the very largest sizes.

This brief review of the development of the refrigerating compressor indicates that recent changes have been rather in the form and construction of the compressor than in any pronounced increase in efficiency. When we come to consider the equipment which makes up the rest of a refrigerating plant we find that changes have taken place, not merely in the construction of the condenser and evaporator, but also in the main principles governing their operation. This applies particularly to the evaporator.

Modern Types of Refrigerating Circuits

Whilst the general principle of the refrigerating cycle has already been explained with reference to Fig. 1, it should be pointed out that most modern refrigerating plants have many refinements added.

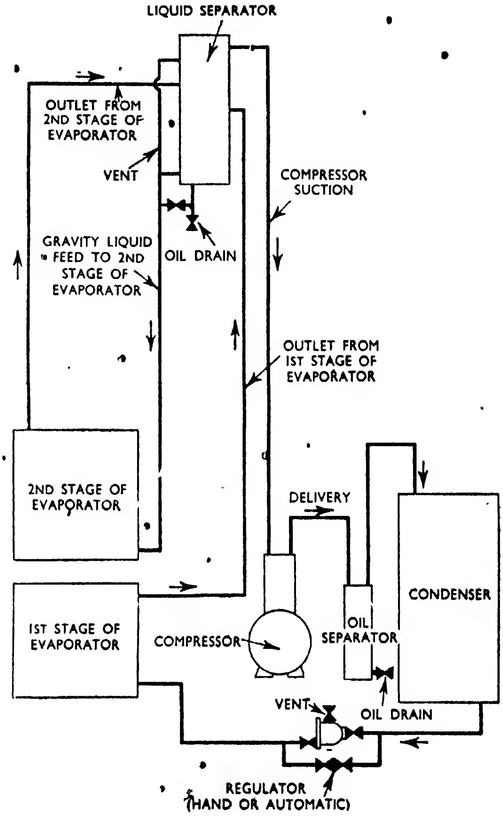


FIG. 10.—CIRCUIT FOR TWO-STAGE EVAPORATION (AMMONIA)

DRY COMPRESSION SYSTEM, WITH HIGH-PRESSURE FLOAT REGULATOR.—Fig. 5 shows the circuit of Fig. 1 modified to operate on the dry compression system. The diagram indicates an automatic regulator of the float type which is usually associated with dry compression, but is not an essential feature. With the arrangement shown, this automatic regulator takes the form of a float which opens an expansion valve as soon as liquid is present. This has the effect of keeping the condenser completely drained of liquid, and no part of the condenser is used for supercooling the liquid after condensation.

The evaporator circuit is provided with a liquid separator consisting of a

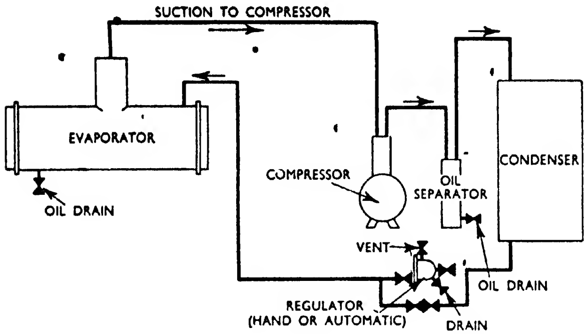


FIG. 11.—CIRCUIT FOR USE WITH BRINE AS COOLING MEDIUM

vessel of such a cross-sectional area that the velocity of the gas flow becomes sufficiently low to allow any free liquid to fall back into the separating vessel and so to return to the evaporator circuit. The gas drawn back to the compressor is thus completely free of entrained liquid, and becomes in almost every case slightly superheated in its passage from the separator vessel to the compressor.

DRY COMPRESSION SYSTEM WITH LOW-PRESSURE FLOAT REGULATOR.—Fig. 6 shows the same circuit arranged with a different type of float regulator which does not necessarily drain the whole of the liquid from the condenser, but maintains a constant level in the evaporator coils. This arrangement is particularly suitable where more than one cooler or other form of evaporator is placed at different levels in the same building.

DRY COMPRESSOR SYSTEM: DIRECT FEED INTO SEPARATOR.—A variation of the circuit is shown in Fig. 7. With this arrangement the liquid passing from the expansion valve is fed directly into the separator vessel instead of to the evaporator coils which, except for the top two or three, are kept completely flooded.

PUMP SYSTEM: HIGH-PRESSURE OR LOW-PRESSURE FLOAT REGULATOR.—The circuits described are used where the evaporator coils form a direct expansion circuit, usually in the form of an air cooler; but where the evaporator coils take the form of grid piping dispersed possibly throughout a number of chambers in a cold store, it is usually necessary to circulate the evaporating liquid through these coils by means of a pump. Figs. 8 and 9 show the circuits adopted in such cases. These may complete either a high-pressure float regulator draining the whole of the liquid from the condenser, or a low-pressure type maintaining a constant level in the liquid separator vessel.

TWO-STAGE EVAPORATION SYSTEM (AMMONIA).—In some of the earliest examples of this system of recirculation of the ammonia liquid, a separator vessel was provided high in the building—sometimes on the roof in an insulated chamber—so that the liquid could be fed by gravity to various expansion circuits as indicated in Fig. 10. The diagram illustrated shows the evaporation carried out in two stages, which is a great convenience in the case of a cold store

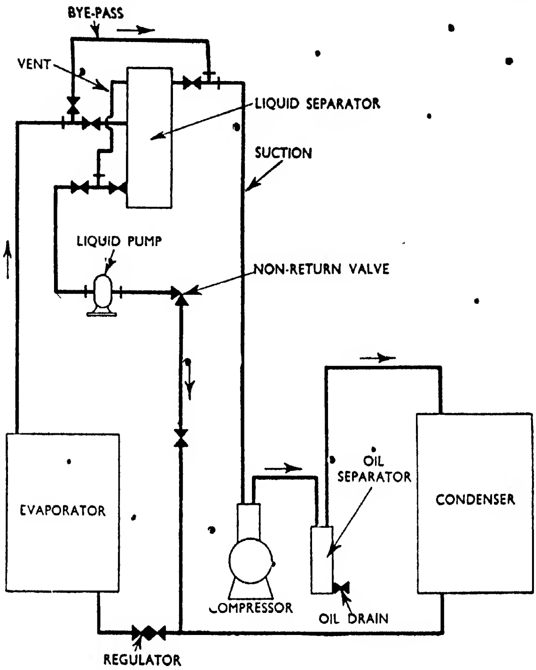


FIG. 12.—MULTIPLE CIRCUITS: PATH OF LIQUID REFRIGERANT

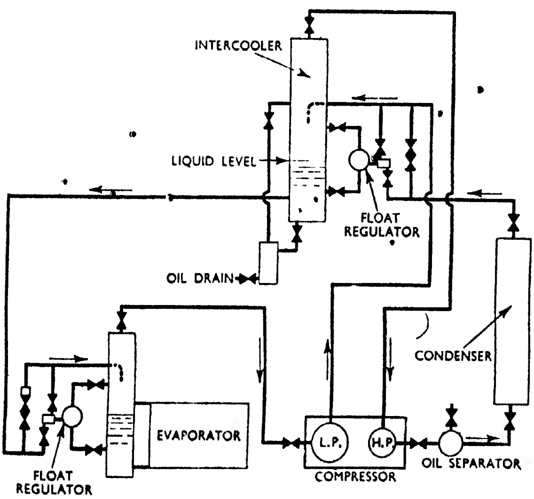


FIG. 13.—CIRCUIT DIAGRAM FOR TWO-STAGE ARRANGEMENT

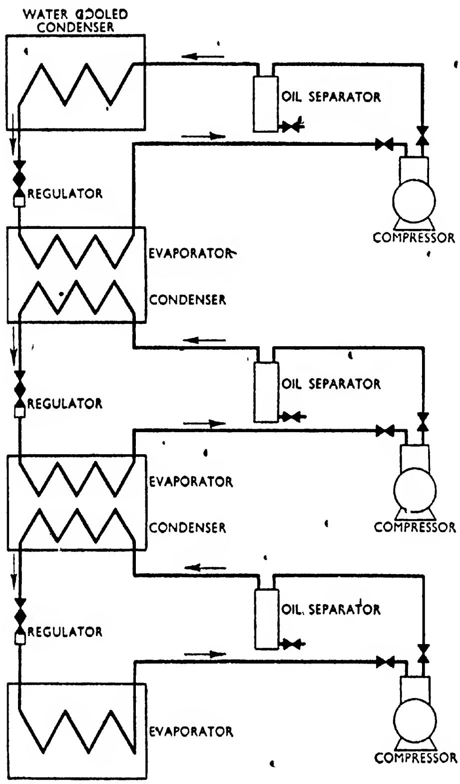


FIG. 14.—CIRCUIT DIAGRAM FOR "CASCADE" ARRANGEMENT

requiring differing temperatures in rooms set aside for different classes of produce.

SYSTEM WITH BRINE AS COOLING MEDIUM. — When brine is used as the cooling medium a somewhat simpler circuit can be used, a very common form being that illustrated in Fig. 11, where the evaporator takes the form of a horizontal vessel containing the liquid ammonia. The brine is passed through tubes held in endplates so that the ammonia boils by absorbing heat from it. In many cases the vessel is fitted with a dome to ensure the separation of any free liquid from the gas passing to the compressor. The form of this evaporator is, indeed, similar to a locomotive-type steam boiler.

MULTIPLE-CIRCUIT SYSTEM: PATH OF LIQUID REFRIGERANT.—In some cases it is found convenient to pass the liquid refrigerant collected in the separator on the suction side to the high-pressure side of the regulator, so that it joins the liquid passing direct from the condenser. This arrangement has been applied with most useful results in cases where the dry compression cycle is to be applied to an existing plant having several circuits spread over a number of freezing chambers or cold stores. The diagram shown in Fig. 12 is typical of this arrangement. A difficulty has to be faced in applying this form of liquid recirculation to an old refrigerating plant, in that the individual circuits on the evaporator side are very frequently much longer than is desirable for recirculation of the liquid, the usual practice in new plants being to use very short individual coil lengths.

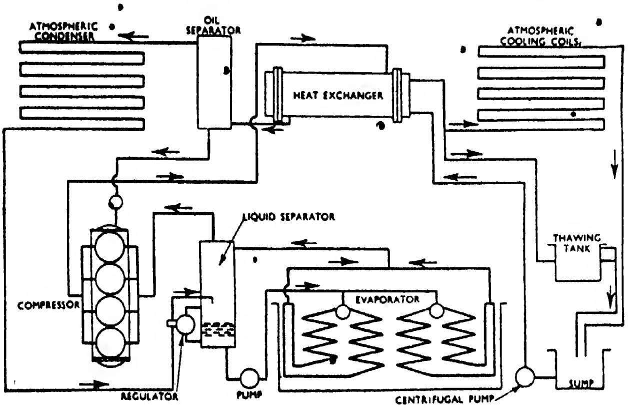


FIG. 15.—CIRCUIT FOR ICE-MAKING PLANT AT FISHING PORT

TWO-STAGE SYSTEM.—When the temperature of evaporation has to be so low that to reach the condensing pressure the ratio of compression becomes greater than is desirable in a single stage, two or more stages of compression must be used. A usual form of circuit using a two-stage reciprocating compressor is shown in Fig. 13. The gas leaving the high-pressure cylinder passes to the condenser, where it becomes liquid and is drained by means of a float valve into the liquid line. This liquid is injected into the delivery from the low-pressure cylinder, and passes with it to the intercooler vessel, where the evaporation of a small portion of the liquid cools the gas on its way to the high-pressure suction.

"CASCADE" SYSTEM.—When the temperature required is so low that it is desirable to use more than one refrigerant, the "cascade" principle may be employed. Fig. 14 is a diagram of such a circuit, using three refrigerants. The highest stage is a normal ammonia circuit with a condenser, cooled by water. The evaporator of this circuit is used for cooling the carbon-dioxide condenser of the second stage, and the carbon-dioxide evaporator in its turn cools the condenser of the lowest stage circuit using ethylene. The evaporator of this last circuit provides the low-temperature cooling required.

Modern Ice-making Plant

An interesting example of the application of the various principles which have been outlined above is shown in the circuit for ice-making plant at a fishing port, illustrated in Fig. 15. From this diagram it will be seen that the gas leaving the compressor passes first through a heat exchanger, in the tubes of which water is circulated and heated; the water is then passed to the thawing tanks, where the heat thus applied thaws the ice from the sides of the cans so that it can be tipped out. Water passing through the heat exchanger is circulated inside these coils when thawing of the ice is not in progress, and is cooled by dock water passing over them.

THE OPERATION AND MAINTENANCE OF MINING MACHINERY

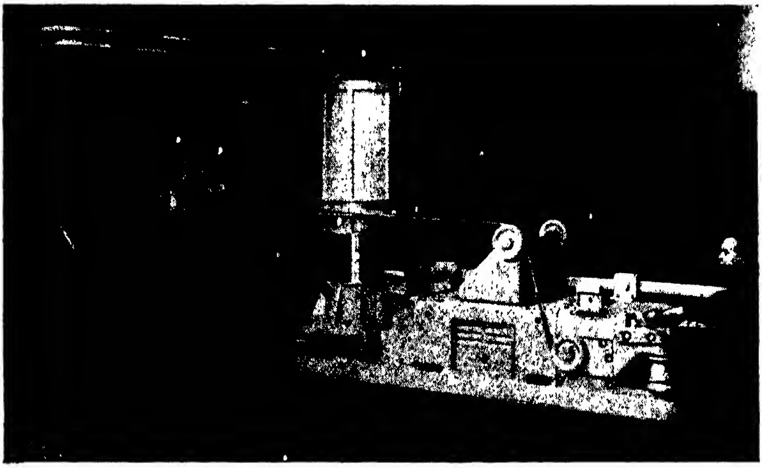


FIG. 1.—TELESCOPIC TURRET OVERHEAD COAL-CUTTER (*Cowlishaw, Walker & Co., Ltd.*)

THE increased application of machinery for the mining of coal in this country has brought a demand for a higher standard of servicing by mechanics, and more intelligent handling by the machinemen. In view of this, it is for mining machinery operators and mechanics that the following maintenance notes have been written.

Most collieries maintain a staff of fitters constantly underground, each one of them detailed to a particular face or district. Certain duties, though, should be carried out by the machinemen themselves. These will be dealt with throughout the article, and the servicing of each class of machine will be found under the relative headings.

COAL-CUTTERS

By far the most common type of coal-cutter in use to-day is the long-wall, either electrically or compressed-air driven. Construction of these involves

FIG. 2.—COMPRESSED-AIR COAL-CUTTER IN ITS THREE UNITS
Showing cutting head, driving unit, and haulage end. (*Mavor & Coulson, Ltd.*)

work of high precision, and working to fine limits. The modern coal-cutter is composed of three main units (Fig. 2), each of which depends upon the other for good performance. They are: cutting end, driving unit, and haulage end. In some machines a smaller section is placed between the motor and cutting end, and houses the gears for running the cutting chain in either direction. Despite the diversity in design, all coal-cutters have one purpose: to cut in the seam or close to it. For this reason it is considered that the cutting end is of more importance than the other sections, although, as mentioned before, each part has an important bearing on the operation of the whole machine. It will be realised that it would be useless to have a good strong motor if the other sections could not stand up to the heavy duty.

The Jib

Severe strain is placed upon the cutting end when at work, and it is important to see that the jib and chain are in good condition. Most jibs are fitted with wearing strips which act as a guide to the chain. These strips should be examined each week and any signs of excessive wear dealt with. A spare jib in good repair is advisable to enable a change over without hindrance to coal

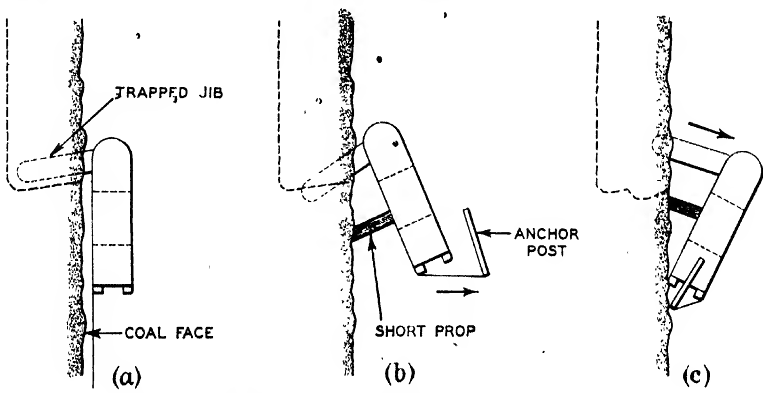


FIG. 3.—EXTRACTING THE JIB WHEN TRAPPED UNDER THE SEAM

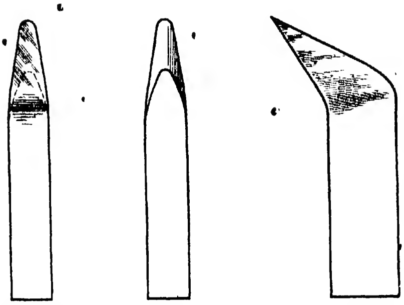


FIG. 4.—CORRECT SHAPE OF CUTTING PICK

output. Warped jibs, usually caused through shot-firing in the vicinity, should be replaced, otherwise trouble may be experienced either with broken pick chains or with the chains coming out of the guides. Shot-firing may also cause fracture of the jib support bar, or jib-stock.

In the event of a trapped jib, the best method of releasing it is as follows. Set the anchor post for the rope and gently pull the haulage end away from the face. Next place a short prop, as shown in Fig. 3, between the machine and the face. Reset the rope again to pull the coal-cutter back towards the coal, leaving the short prop to act as a fulcrum, and lever the jib out. It may be necessary to run the picks during the operation, but all personnel except the operator must keep clear.

Pick Chains

Pick chains kept to the correct tension will last much longer and will reduce wear on the jib. It is a good precaution to adjust the strap-link type of chain to approximately 1½ in. play at a point midway between the sprocket and jib entry. Other kinds of chain, however, should be set to the makers' specifications. The amount of wear of the chain has a bearing upon the useful life of the chain sprocket. Any excessive end-play between links and boxes will result in the chain not engaging properly with the sprocket teeth, and may eventually lead to a fracture of either the chain, sprocket, shaft, bearings, or all four.

It is essential to have the picks correctly shaped and sharpened. Not only do they cut better, but they then reduce the load on the motor and gears. The front of the pick must be wider than the back shoulder to give cutting clearance. A rounded tip of about $\frac{1}{16}$-in. radius usually gives the best results. This is shown in Fig. 4.

Setting the Picks in Position

Special tipped picks are used in many cases, and they cannot be sharpened in the usual way. The maker's instructions should be followed, for different types need different treatment; some are ground by a tool provided for the purpose.

Speed in cutting is of no value if only half of the jib is in use. The ideal is to have it well into the coal, and level. Should the picks not be correctly set or sharp, there is every possibility of the jib rising or digging into the floor. Every

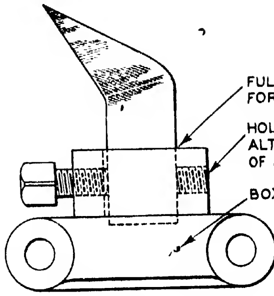


FIG. 5.—CORRECT METHOD OF INSERTING PICKS INTO PICK CHAIN BOXES (NOT TO SCALE)

pick must therefore be set to project the correct distance from its pick-box. A gauge is usually provided for setting the picks quickly and accurately. Any tendency for the jib to rise in the seam can, in most cases, be rectified by the simple operation of extending the bottom picks about $\frac{1}{8}$–$\frac{1}{4}$ in. beyond the others. This guarantees extra clearance for the bottom of the jib, allowing it to regain its correct level. Digging into the floor can be overcome by a similar adjustment to the top picks. If at any time it is found essential to put packing under the body of the coal-cutter, first ensure that the machine is fitted with a full-length sledge that extends well under the chain and sprocket. If the sledge is only three-quarters length there is danger in the material used fouling the chain and causing accidents. Always let the motor come to a standstill before engaging the gears, otherwise there is a possibility of damaging the teeth.

Lubrication

Lubrication is a very important factor in determining the useful life of any machine, but it is essential that the correct grade of oil specified by the maker be used. Over-lubrication should be avoided in the case of coal-cutters, as any excess may find its way into the motor chamber. In the case of electrically driven machines the result may be a burn-out of the windings. Cutters driven by air require a special oil to prevent freezing at the exhaust ports, and to give ample lubrication to the rotors. Some typical charts for the lubrication of coal-cutting machines are given on pages 508–511.

It should also be noted that the hose supplying the compressed air must be cleaned out before it is connected to the machine. Blowing it out prevents any dust or other matter from reaching the turbine. After cutting the face with a compressed-air machine, the rotors should be given a coating of light oil to keep them free from rust. All filters in the air intake must be kept clean and clear of even the slightest obstruction. A lever must on no account be used on the rotor teeth to turn the rotors, however important it may be to free them. Oiling the cutter chain at the start of every cutting shift improves the machine's performance and increases the life of the chain and jib.

Coal-cutter Switchgear

The majority of coal-cutters are electrically driven, and a few words on the switchgear will be of interest to the machinemen.

Never insert or remove the plug without first isolating the cable at the panel switch. Keep the plugs clean. Move the switch decisively from step to step, with

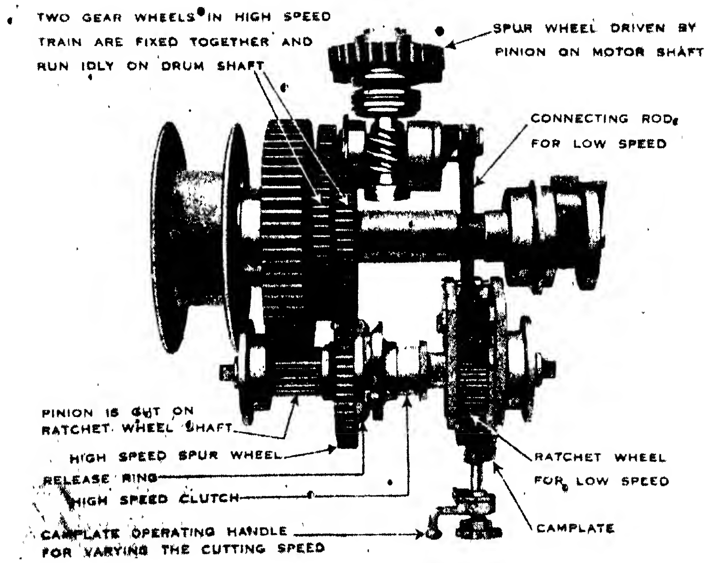


FIG. 6.—MOVING PARTS OF THE HAULAGE GEAR
Right-hand rope drum is removed to show hub and pawl. (*Mavor & Coulson, Ltd.*)

a short pause in each position. Do not switch off a remote-control switch all the way in one quick action, but pause for a moment in the intermediate notch. If this pause is neglected, the switch on the machine may beat the contactor at the gate end, breaking the current at the isolating contacts of the switch and burning them unnecessarily. Arcing at contacts pits the copper surfaces, deposits a semi-conducting film on the insulation, and causes a certain amount of moisture to settle on the switch-chamber casing, thus promoting corrosion.

If any difficulty is experienced in starting the motor the machineman should not interfere with the starter but should report the fact to the electrician. The starter should be inspected by the electrician at regular intervals, usually once a fortnight, and the contacts cleaned and adjusted as necessary. The connections in the pilot circuit must be good, the panel switch kept in first-class order, and the voltage steady.

Haulage Gear

The haulage section of the modern coal-cutter (Fig. 6) is a good example of fine engineering, and although built to fine limits, it is extremely powerful; but to maintain this power the gear requires regular attention. The haulage end must be kept clean, and in some makes of coal-cutter the driving and retaining

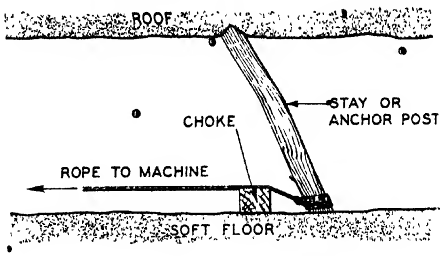


FIG. 7.—ANCHOR-POST SETTING FOR A SOFT FLOOR

pawls should be examined weekly for any signs of chipping or blunting of the edges; any that are found defective should be replaced with new ones. The same attention must also be given to the ratchet wheel. Weak or broken pawl springs require replacement, or serious damage will be caused to the teeth of the expensive ratchet wheel. Back lash in the action of the cam-operating levers should be taken up, otherwise the correct engagement of the feed to the haulage will not be possible. A regular check of the bearings will ensure that the train of gears all mesh properly.

Some machines are now fitted with a clutch to prevent the possibility of haulage rope breakage in addition to relieving the gears of strain. These clutches must be set to the correct limits as recommended by the maker, and must not be interfered with by inexperienced persons. The adjustment should be made by using a dynamometer, the usual setting being in the region of a 10,000–15,000 lb. pull on the rope.

The use of frayed ropes for the haulage is a dangerous practice, and these should be replaced with new ones. Serious accidents to the hands of the operators can be avoided by a careful watch on the ropes, particularly where they are fixed to the anchor post. The coiling of the rope on to the drum should be done evenly, and without kinking. This makes the work of obtaining slack rope much easier, as well as saving valuable time. On most coal-cutters there are two rope pulleys, termed bollards, and they require oiling at regular intervals to give a free working of the rope on to the haulage drum.

Anchor posts must be set to the same line as the direction of the rope pull, at a slope of about 45°, with the top of the post well set into the roof. If the floor is of a soft nature, it is an advantage to place a choke of hardwood under the rope and immediately to the front of the post (Fig. 7).

FACE CONVEYORS

Broadly speaking, only two types of conveyor are used for face duty: they are the belt and the chain. Both are good, but the chain has a wider field of application, it is more suitable for retarding with the gradient or conveying uphill, and under wet conditions is more durable than the belt.

Grading of the Belt

All types of conveyors require a good straight line and correct grading. Rapid wear of the belt can be expected if the above points are neglected, and frequent breakages will no doubt occur during coal work. Bottom loading

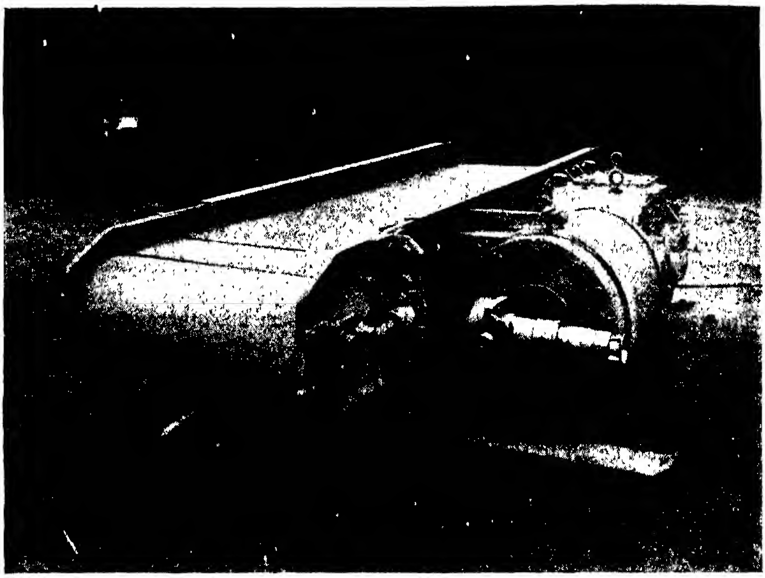


FIG. 8.—A 25–35-H.P. FACE-BELT CONVEYOR DRIVE HEAD
(*Cowlishaw, Walker & Co., Ltd.*)

belts are in wide use, especially in thin seams which are being worked to-day, but the system of application is often very inefficient. Although they are usually supplied by the makers, rollers and supports are seldom used, with the result that the belt scrubs along the floor under heavy load. It is obviously illogical to lubricate the return drum and the snub rollers of the drive head, and yet ignore the serious friction on the belt under such conditions.

The inclination of the belt from the drive head should be gradual, and without any sudden dip. This rules out any tendency for the belt to lift when it approaches the discharge end, or when it is fully loaded. It should be mentioned that grading is more important where top loading belts are employed. Ploughs fitted to bottom belts must be securely fastened, and at such an angle that the coal is delivered to the centre of the gate conveyor, if one is used.

The Driving Unit

A form of semi-flexible coupling is usually interposed between the motor and gearcase of the bottom belt conveyor, and it requires a fair amount of attention (Fig. 8). Even though the motor may have been set into perfect alignment during the installation, the movement of the floor may have upset the original setting, and a careful watch must be kept on it. Replacing the rubbers or other shock-absorbing material is much cheaper than renewal of the couplings.

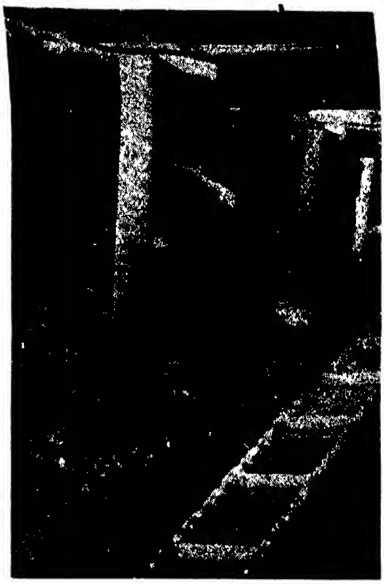


FIG. 9.—CHAIN CONVEYOR AT THE COAL FACE (*Cowlishaw, Walker & Co., Ltd.*)

Lubrication of the driving unit should be attended to at the beginning of every shift and lubricators should be topped up to the correct level. Avoid spilling oil on the belt, as it may cause surging on the drums, and will in any case rot the rubber. The bearings of the return drum also need greasing at regular intervals, largely on account of their being subject to the penetration of small coal into the bearing housings.

It is essential that the ends of the belt be cut perfectly square when remaking a joint, otherwise the belt will "wander" and rub against the steel edges of the structure. For face conveyors the use of hinge joints is not satisfactory, as the radius of the driving and return drums is too short to allow the amount of flexibility required. Certain types of joint are available that lend themselves to follow even the smallest-diameter drum without any danger to tearing of the belt.

It is not intended to discuss the actual moving over of a face conveyor into the new track, as the conditions vary throughout the whole industry, and the nature of both floor and roof each have a bearing on the method used.

Chain Conveyors

The use of chain conveyors for longwall duty is finding favour at a great number of places, largely because of their robust construction and ability to deal with large output without spillage. Grading, however, is of foremost importance, and must be attended to, to prevent any possibility of the chain riding above the coal. The chains, usually in lengths of 6 ft., are joined together by either bolt and nut or pins and cotters. For face work the pin and cotter is quicker to insert and remove, but the cotter pins must be put in and the tails opened to secure them. The heads of the coupling pins should always be on the outside, except where the makers state otherwise. The tension of the chain is a matter of great importance, and should be adjusted so that about 4 in. of sag is available on the underside of the sprocket in the case of a head end drive. With a tail drive, such as a retarder, the slack chain is given off at the top and requires setting so that there is no doubling back of the chain links. A good feature of the tail-driven chain conveyor is its ability of following the contours of an undulating floor, and is worth considering when dealing with the problem of floor lift.

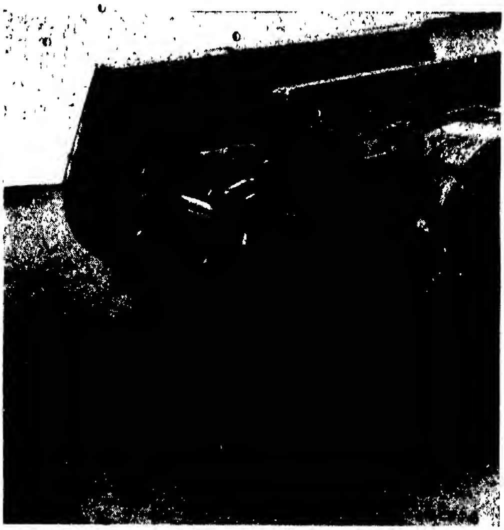


FIG. 10.—SHEARING-PIN DEVICE ON CHAIN-CONVEYOR DRIVE HEAD
(*Cowlishaw, Walker & Co., Ltd.*)

Steel props should not be set on the drive head, as this not only strains the bearings and shafts, but may lead to a fracture of the gearcase. Special feet are, in most cases, provided for the staying of the head, and must be used at all times.

GATE AND TRUNK BELT CONVEYORS

Prior to the installation of any major conveyor system, it is desirable to have the roadway surveyed, and a line clearly marked on the roof. No matter how this line is to be followed, either as the centre of the belt, or as a guide to the structure; there must be sufficient side clearance for a supply roadway. On the other hand, the space at the back of the structure should be such that any large pieces of coal that fall from the belt will not rub against the belt edges.

Installation

It is important to see that the drive head (Fig. 11) is set on to a solid foundation, and is perfectly level throughout. Always make certain that the clearance for the top belt is sufficient to allow easy passage to the largest piece of coal that is likely to be carried without fouling the roof supports. The positioning of the head pulley should be determined by the speed of the belt, so as to avoid overthrowing or short pitching, which would create spillage and extra work, and which would also reduce the rate of tub filling.

The Loop Take-up Section

The addition of a loop take-up section is a good feature in keeping the number of joints in the belt to a minimum. The loop frame must be set to the same level and plane as the drive head, to ensure a straight run of the belt and to ease the work of adjustment. The adding of structure to the frame must be carried out with consideration to the grading, and a fall of 4 in. to each 8 ft. or 9 ft. of section is a good average. All sections must be cross levelled, and kept

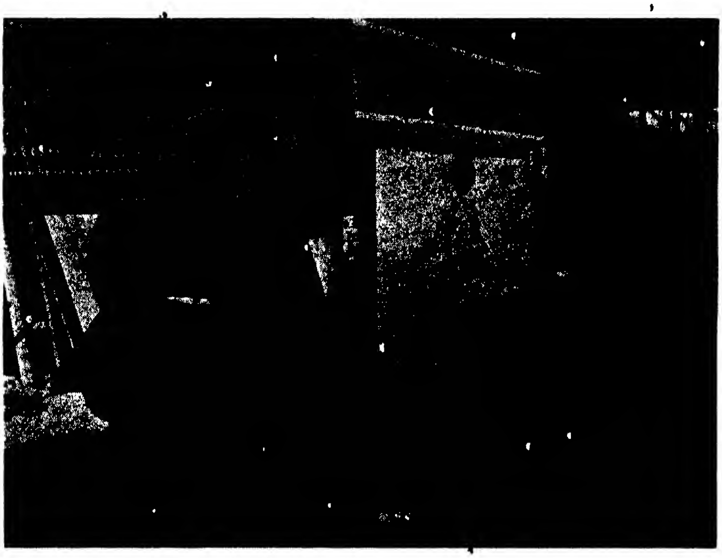


FIG. 11.—A 75-H.P. TRUNKBELT DRIVE HEAD, SHOWING HYDRAULIC COUPLING (*Cowlishaw, Walker & Co., Ltd.*)

dead to line. It is a great advantage to have the structure well up from the floor to allow easy examination of the return belt.

Threading of the belt through the drive is best done by starting from the bottom. Have the loop sliding drum well forward so that the slack can be taken up without resorting to cutting the belt. During the first run, do not start immediately to adjust the idler rollers, but give the belt time to settle down. Begin any adjustment necessary by paying first attention to the return belt. This is of more importance than the top because the bottom one is liable to constant friction by contact with the structure. Work in towards the tension end and train the return idlers, and on the way back to the driving unit the top ones can be set. A point to remember is that any adjustment to an idler does not affect the belt at that point but always some two or three yards farther on.

Troughed Idlers

In the event of a troughed idler refusing to revolve, have it removed, as the friction to the belt will cause it to run out of line. It is the standard practice now for the manufacturer to construct the troughed idlers with the outside ends of the side rollers a little in advance of the centre one. The idea of this is to train the belt, and it is essential that they be mounted the correct way, as shown in Fig. 13. Self-aligning rollers that pivot from the centre of the stand should be mounted at intervals of 50 yards or so. Troughed rollers less than 4 in. in

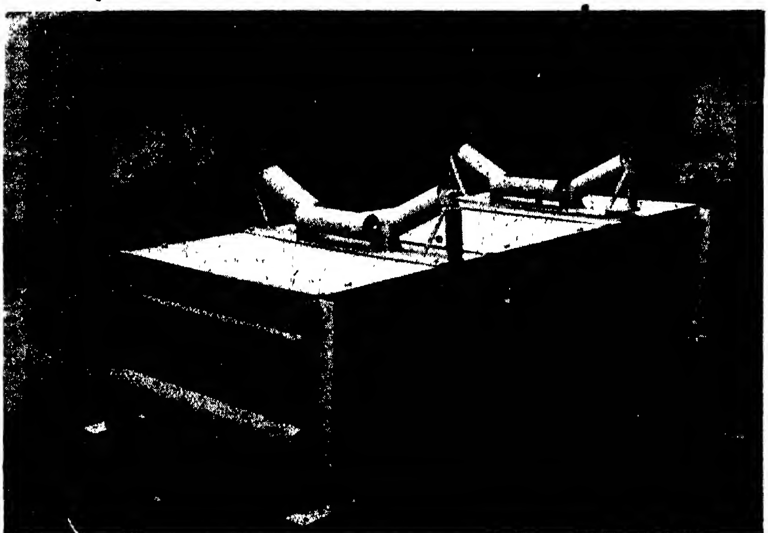


FIG. 12.—A 36-IN. TRUNK CONVEYOR, INTERMEDIATE SECTION
(*Cowlishaw, Walker & Co., Ltd.*)

diameter should not be used on a belt that travels at 200 or more feet a minute. The speed is rather high for small rollers, and will result in the collapse of the bearings.

Cross-levelling of Conveyor Structure

Cross-levelling of the conveyor structure must be carried out from end to end of the whole lay-out, and a spirit level used on the rollers to verify it.

In all cases it is desirable for the conveyor to be kept clean, but if the line has been followed diligently, and care taken with the levelling, the need for cleaning up will not be so frequent.

Protection of Tension Ends

Tension ends are often subject to much abuse by the stone work in the ripping or brushing shifts, and to prevent damage to them, a guard of steel plate should be made to protect them. This device is most effective when semi-circular in shape, as the stone usually glances off without displacing it. A suitable guard would be about 8 ft. long and wide enough to straddle the conveyor. It should also be fitted with skids to facilitate installation.

Chain Scraper-feeder

The adopted method now of loading on to the gate belt is to employ a chain scraper-feeder. This can be driven by a small motor and gearbox, or by the belt

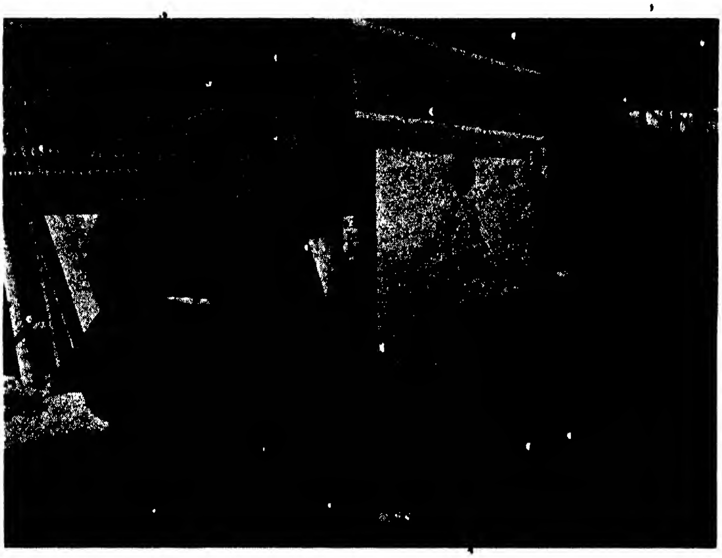


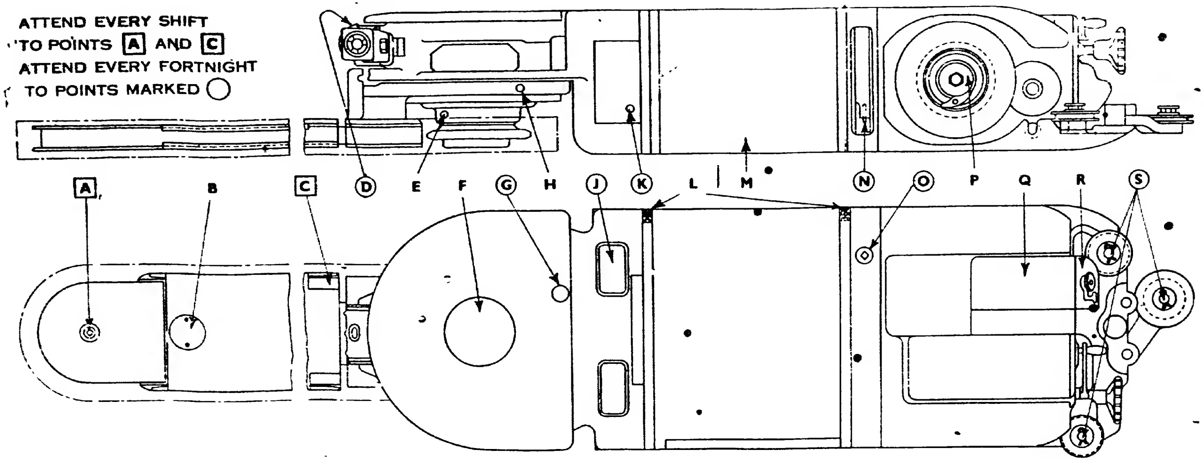
FIG. 11.—A 75-H.P. TRUNKBELT DRIVE HEAD, SHOWING HYDRAULIC COUPLING
(*Cowlishaw, Walker & Co., Ltd.*)

dead to line. It is a great advantage to have the structure well up from the floor to allow easy examination of the return belt.

Threading of the belt through the drive is best done by starting from the bottom. Have the loop sliding drum well forward so that the slack can be taken up without resorting to cutting the belt. During the first run, do not start immediately to adjust the idler rollers, but give the belt time to settle down. Begin any adjustment necessary by paying first attention to the return belt. This is of more importance than the top because the bottom one is liable to constant friction by contact with the structure. Work in towards the tension end and train the return idlers, and on the way back to the driving unit the top ones can be set. A point to remember is that any adjustment to an idler does not affect the belt at that point but always some two or three yards farther on.

Troughed Idlers

In the event of a troughed idler refusing to revolve, have it removed, as the friction to the belt will cause it to run out of line. It is the standard practice now for the manufacturer to construct the troughed idlers with the outside ends of the side rollers a little in advance of the centre one. The idea of this is to train the belt, and it is essential that they be mounted the correct way, as shown in Fig. 13. Self-aligning rollers that pivot from the centre of the stand should be mounted at intervals of 50 yards or so. Troughed rollers less than 4 in. in



LUBRICATION OF UNDERCUTTING MACHINE (WITH ELECTRIC MOTOR)

After filling the gear chambers, lubricate as follows:

Once a shift

A (sprocket jib only)—Withdraw the plug, fill the recess with heavy oil or light grease, and screw the plug home

C—Oil the cutter chain

Once a fortnight

D—See that gummer gearcase is full

G—Pour in half a gallon of gearcase oil

J—Remove the cover and see that the lowest gearwheel dips in oil, replenish if necessary

K, *N*—Make sure that the overflows are open. There are two on each side of the machine

O—Fill the chamber to level with top of machine

S—Oil the three pulleys

Once a month

B—Take off the cover and pack the recess with grease

R—Take off the cover and pour a little gear oil on to the parts carried in the cover

Once every two months

Use the grease gun to inject grease into the following nipples:

E, sprocket shaft bearing—$\frac{1}{2}$ oz.

H, jibhead bearing and seal—1 oz.

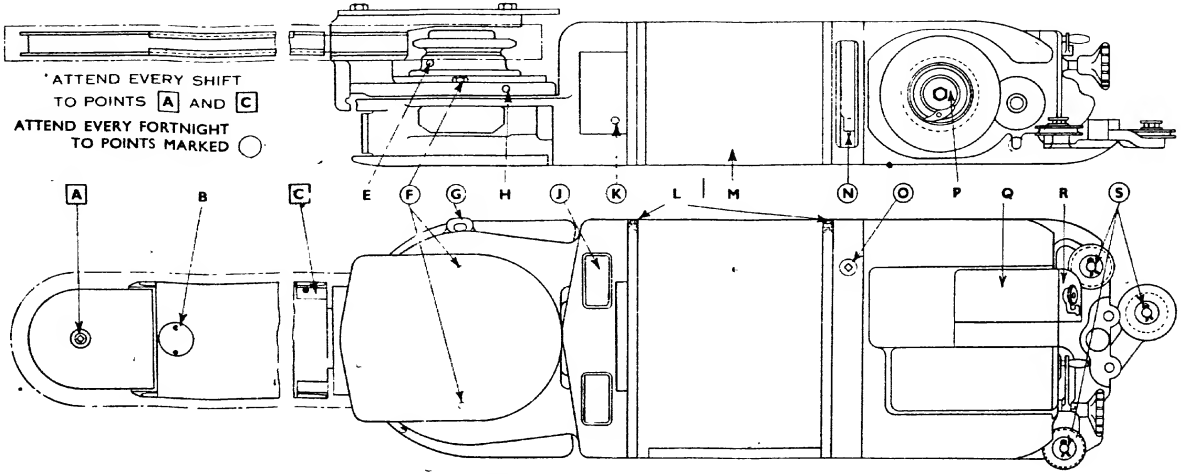
L, two motor bearings—$\frac{1}{4}$ oz. each. See note below

F, *P*—Take off covers and pack the recesses with grease

M—Starter chamber cover: remove the cover to see that the motor casing is free from oil

Note.—Points *L* should be greased, if at all possible, while the machine is still warm

(*Mayor & Coulson, Ltd.*)



LUBRICATION OF OVERCUTTING MACHINE (WITH ELECTRIC MOTOR)

After filling the gear chambers, lubricate as follows:

Once a shift

A (sprocket jib only)—Withdraw the plug, fill the recess with heavy oil or light grease, and screw the plug home

C—Oil the cutter chain
E—Grease the nipple by means of gun
F—Oil through either of the two plugs
H—Grease the nipple by means of gun

} *Every shift, in abrasive holing; weekly, when cutting coal*

Once a fortnight

G—Pour in half a gallon of gear oil

J—Remove the cover and see that the lowest gearwheel dips in oil, replenish if necessary

K, N—Make sure that the overflows are open. There are two on each side of the machine

O—Fill the chamber to level with top of machine

S—Oil the three pulleys and any other pulleys on the baseplate. Screw jacks. If the machine is mounted on jacks, once a fortnight inject ½ oz. of grease into each jack-screw housing

Once a month

B—Take off the cover and pack the recess with grease

R—Take off the cover and pour a little gear oil on to the parts carried in the cover

Once every two months

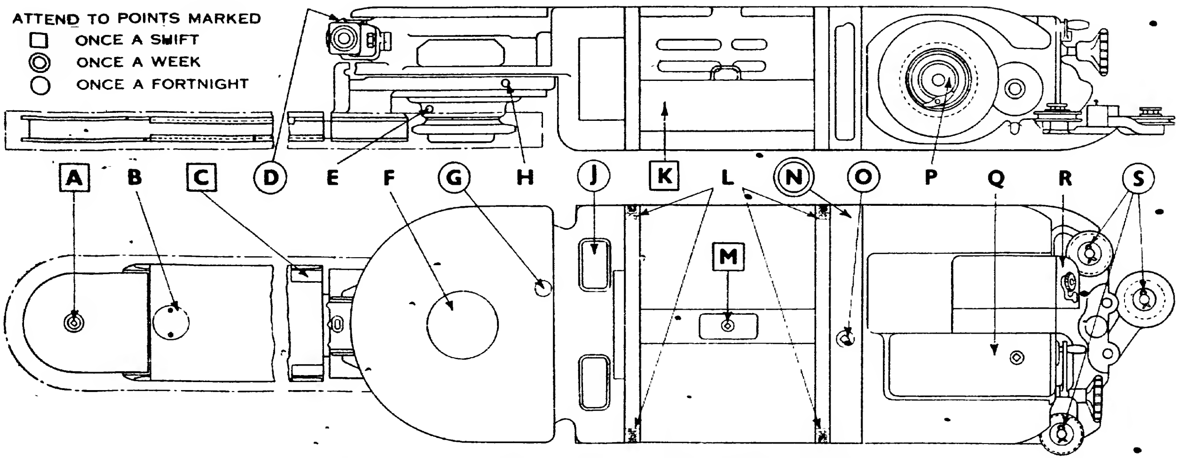
L—Apply grease gun to inject ½ oz. of motor grease into each of the two motor bearings. See note below

M—Starter chamber cover: remove the cover to see that the motor casing is free from oil

P—Take off the covers and pack the recesses with grease

Note.—Points *L* should be greased, if at all possible, while the machine is still warm

(*Mavor & Coulson, Ltd.*)



LUBRICATION OF UNDERCUTTING MACHINE (WITH AIR TURBINE)

After filling the gear chambers, lubricate as follows:

Once a shift

A (sprocket jib only)—Withdraw the plug, fill the recess with heavy oil or light grease, and screw the plug home
C—Oil the cutter chain
K—Raise the doors at end of shift to keep out dust
M—Fill with anti-freezing oil to level with top of siphon tube

Once a fortnight

D—See that gummer gearcase is full
G—Pour in half a gallon of gear oil
J—Remove the cover and see that the lowest gearwheel dips in oil; replenish if necessary
O—Fill the chamber to level with top of machine
S—Oil the three pulleys

Once a month

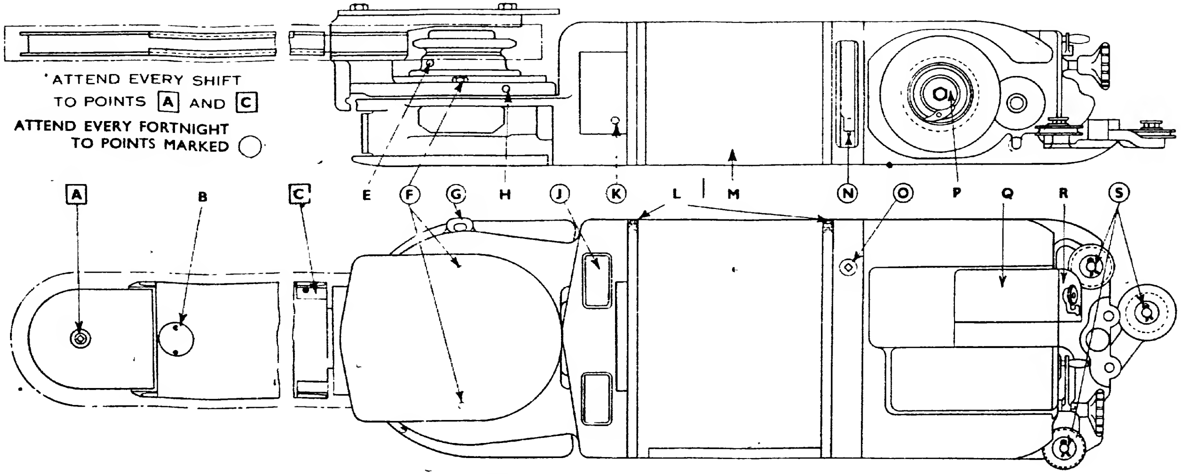
B—Take off the cover and pack the recess with grease
R—Take off the cover and pour a little gear oil on to the parts carried in the cover

Once every two months

Use the grease gun to inject grease into the following nipples:

E, sprocket-shaft bearing
H, jibhead bearing and seal
L, four rotor bearings
F, *P*—Take off the covers and pack the recesses with grease

(*Mavor & Coulson, Ltd.*)



LUBRICATION OF OVERCUTTING MACHINE (WITH ELECTRIC MOTOR)

After filling the gear chambers, lubricate as follows:

Once a shift

A (sprocket jib only)—Withdraw the plug, fill the recess with heavy oil or light grease, and screw the plug home

C—Oil the cutter chain
E—Grease the nipple by means of gun
F—Oil through either of the two plugs
H—Grease the nipple by means of gun

} *Every shift, in abrasive holing; weekly, when cutting coal*

Once a fortnight

G—Pour in half a gallon of gear oil

J—Remove the cover and see that the lowest gearwheel dips in oil, replenish if necessary

K, N—Make sure that the overflows are open. There are two on each side of the machine

O—Fill the chamber to level with top of machine

S—Oil the three pulleys and any other pulleys on the baseplate. Screw jacks: if the machine is mounted on jacks, once a fortnight inject ½ oz. of grease into each jack-screw housing

Once a month

B—Take off the cover and pack the recess with grease

R—Take off the cover and pour a little gear oil on to the parts carried in the cover

Once every two months

L—Apply grease gun to inject ½ oz. of motor grease into each of the two motor bearings. See note below

M—Starter chamber cover: remove the cover to see that the motor casing is free from oil

P—Take off the covers and pack the recesses with grease

Note.—Points *L* should be greased, if at all possible, while the machine is still warm

(*Mavor & Coulson, Ltd.*)

The machines are usually of two types, air cooled or water cooled, and either single or two-stage. Single-stage compressors have one or two cylinders of equal bore and stroke, while the two-stage ones have two or three, with one cylinder of smaller bore than the other two. This small one is the high compression cylinder, from which the air is forced into the receiver. Two-stage compressors invariably have an intercooling section between the two stages, which takes the shape of a radiator composed of a large number of small pipes through which the hot air is passed. Behind the radiator a fan revolves, usually driven direct from the engine crankshaft, and it is of vital importance that it turns in the right direction.

Compressors driven by V-belts should be placed so that the driving belts are at the bottom of the pulleys. This is to maintain a good arc of contact even after the belts have stretched a little.

The valves in the engine should be examined every month, and deposits of carbon removed from the cylinder heads. The provision of a spare set of valves is sound practice, and will enable faulty ones to be replaced without any serious delay. In the event of a fusible plug blowing, check over the valves in the high-pressure head for leakage. A blow from the safety valve of the intercooler often means a fault has developed in the heads of the low-pressure stages.

The small cut-out valve is of delicate construction, and is usually set to give a time lag of 5–6 lb. on the pressure gauge. For example, assuming that it is set to cut off at 60 lb., and upon reaching this figure the valve comes into action causing the compressor to "freewheel," as the air is used from the receiver, the pressure gradually falls to 55 lb. and the cut-out regains its former position, thereby allowing the engine to begin compressing again. The adjustment to this time lag is, in most cases, made by the use of thin brass shims in the body of the cut-out valve, and should only be carried out by experienced persons.

Air receivers should be drained of water at frequent intervals, and the safety valve tested each day by lifting it from its seat with a short lever. This also keeps the face of the valve clean.

The pipes, cocks, and pump of water-cooled engines must be kept in clean condition, and the scale removed from the water jackets of the cylinder heads. A ½-in. cock attached to the air discharge pipe and a short hose are well worthwhile, as they are useful for cleaning out the intercooler and air-intake filter.

There are a great many more machines of various kinds in the coalmines to-day; the types dealt with are most common, and to attempt to describe each one separately would require much more space than is here available. It is hoped, however, that the advice offered in the space of this article will be of real help to those who are engaged in the operation and servicing of mining machinery.

J. H. S. Y.

CLASSIFIED KEY

MACHINE TOOL PRACTICE

WELDING AND METAL-CUTTING

SHEET AND PLATE METAL WORK

FORGING AND PRESSWORK

PRODUCTION PLANNING

ENGINEERING PRODUCTION WORK

GAS, OIL, AND PETROL ENGINES

STEAM ENGINES, TURBINES, AND BOILERS

POWER TRANSMISSION

OPERATING ENGINEERS' WORK

INDEX TO VOLUMES I—IV

NOTE: The italic numerals indicate the numbers of the volumes.